AD-A031 744 MASSACHUSETTS INST OF TECH CAMBRIDGE DEPT OF NUCLEAR--ETC F/G 13/1
DESIGN OF A NUCLEAR POWERED TOTAL ENERGY SYSTEM FOR FORT BRAGG,--ETC(U)
MAY 76 J W STETKAR, F R BEST, M W GOLAY DAAK02-74-C-0308

UNCLASSIFIED USAFESA-RT-2011 NL

1 OF 4
AD
A031744

REPORT FESA-RT 2011

DESIGN OF A NUCLEAR POWERED TOTAL ENERGY SYSTEM FOR FT. BRAGG, NORTH CAROLINA

NUCLEAR ENGINEERING DEPARTMENT
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
CAMBRIDGE, MASSACHUSETTS

31 May 1976

FINAL REPORT

APPROVED FOR PUBLIC RELEASE - DISTRIBUTION UNLIMITED

PREPARED FOR: US Army Facilities Engineering Support Agency
Fort Belvoir, Virginia 22060

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER FESA-RT 2011	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) Design of a Nuclear Powered Total Energy System for Ft. Bragg, North Carolina		5. TYPE OF REPORT & PERIOD COVERED FINAL REPORT
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) John W. Stetkar Frederick R. Best M. W. Golay		8. CONTRACT OR GRANT NUMBER(s) DAAK02-74-C-0308
9. PERFORMING ORGANIZATION NAME AND ADDRESS Massachusetts Institute of Technology Nuclear Engineering Department Cambridge, Massachusetts		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS 6.27.19-4A762719AT41-T6-005
11. CONTROLLING OFFICE NAME AND ADDRESS US Army Facilities Engineering Support Agency Fort Belvoir, Virginia 22060		12. REPORT DATE 31 May 76
		13. NUMBER OF PAGES 305
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		15. SECURITY CLASS. (of this report) UNCLASSIFIED
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

Eont (9) (6) (10) (15) (11) (12) 306 p.

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR ~~PUBLIC RELEASE - DISTRIBUTION UNLIMITED.~~

(18) USAFESA-RT

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

(19) 2011

18. SUPPLEMENTARY NOTES

Prepared in association with the Energy Laboratory of the Massachusetts Institute of Technology, Cambridge, Massachusetts.

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

ENVIRONMENT; ENVIRONMENTAL IMPACT; NUCLEAR; TOTAL ENERGY; TOTAL UTILITY; POWER PLANTS; HIGH TEMPERATURE GAS COOLED REACTOR; PRESSURIZED WATER REACTOR; HOT WATER DISTRIBUTION SYSTEMS.

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A Total Energy System (TES) is designed to supply the thermal and electrical energy requirements of Fort Bragg, North Carolina for a period of 30 years, with startup scheduled for early 1985. Considered for use as the central station power plant for this system are a combined coal gasification, fossil-fired gas turbine (CGGT) power plant and a direct Brayton cycle high-temperature gas-cooled reactor, helium gas turbine (HTGR/GT) power plant. Several utility system configurations affording different thermal/electrical energy demand ratios are studied for each supply option. With the primary system (continued

next page

DD FORM 1473, 1 JAN 73 EDITION OF 1 NOV 65 IS OBSOLETE UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

401186

1B

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

optimization criterion being the choice of the TES providing a minimum of total energy costs over the system lifetime, it is found that the optimal thermal/electrical load split for each supply option occurs at approximately 75% of the base's total energy demands supplied thermally. Within the limits of the unit-cost assumptions made and for the range of cases studied, it is found that the present-worth total cost of the optimized HTGR/GT system (in 1985 dollars) is $245.7 million and the corresponding optimal system cost for the fossil CGGT alternative is $181.7 million. Further studies are recommended to investigate the sensitivity of this 26% cost differential to variations in the mode of power plant operation and to design modifications in the thermal energy distribution piping network.

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

TABLE OF CONTENTS

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS(continued)

TABLE OF CONTENTS (continued)

LIST OF FIGURES

LIST OF FIGURES (continued)

LIST OF FIGURES (continued)

LIST OF FIGURES (continued)

LIST OF FIGURES (continued)

LIST OF TABLES

LIST OF TABLES (continued)

LIST OF TABLES (continued)

LIST OF TABLES (continued)

CHAPTER 1

INTRODUCTION

1.1 Foreword

This is the final report under a contract between the Massachusetts Institute of Technology and the United States Army Corps of Engineers to develop a conceptual design for a Total Energy System (TES) supplying both electrical and thermal energy to large U.S. Army bases. The system discussed in this report is a second iteration optimization of the design for a 1985 Total Energy System for Ft. Bragg, North Carolina. Use of both nuclear, fossil-fueled and hybrid (nuclear plus fossil) power stations are considered as well as the dependence of power station costs upon the thermal/electrical apparatus mix in the customer sector. The sensitivity of TES costs to changes in capital costs, fuel costs and Thermal Utility System (TUS) cost is also presented. Recommendations are made regarding the optimum TES for Ft. Bragg.

It is found that a minimum cost Total Energy System for both the nuclear and fossil options occurs when the thermal/electric space conditioning split is set at 75%/25%. Additionally, it is shown that for the fossil fired plant to remain less expensive than the nuclear option, the projected cost of coal must remain less than $52/ton averaged over plant lifetime.

1.2 Background

During the past ten years, oil and natural gas have supplied 75% of the nation's energy needs, with coal supplying 21% and all other energy sources, including nuclear, accounting for only 4% of the total. [1] Oil and gas have been the preferred energy sources because they were easily obtained, transported and converted to electrical and thermal energy. Recently however, the scarcity of natural gas and the rising cost of foreign, interruptible oil supplies has lead to consideration of alternative energy sources for meeting energy demands. Solar power, wind power, geo-thermal, fusion and many other energy sources are being investigated and developed to meet national energy needs. However, coal and nuclear power are the prinicpal competitors in the current energy market place. Each fuel has its own characteristic advantages and disadvantages, some of which are listed in Table 1.1.

As is seen in Table 1.1, there is no decisive factor which would lead to choosing one energy source over the other. In the report prepared by Metcalfe and Driscoll, "Economic Assessment of Nuclear and Fossil-Fired Energy Systems for DOD Installations," [2] nuclear plants and fossil-fired gas turbine plants are shown to be economically competitive in the size range of interest (50-100 MWe). Metcalfe, et al., considers pressurized water reactors (PWR), high temperature

TABLE 1.1

NUCLEAR VERSUS COAL PLANT CHARACTERISTICS

Nuclear	Coal
1. Complex licensing procedures and operating requirements.	1. Can be operated and maintained by fewer and less-well-trained personnel than a nuclear unit.
2. High capital cost	2. Lower capital cost
3. Low fuel cost	3. High fuel costs
4. Several years (3-6) of operation on a single fueling	4. Impractical to store more than a few months fuel supply on site
5. Low environmental impacts	5. Meeting exhaust emission standards imposes large economic penalties
6. Low risk, but high consequence reactor safety hazards exist	6. Can be located closer to load center
7. Requires relative isolation of the plant (exclusion area)	7. Airborne chemical emissions impose significant public health risks
8. Cooling towers required for dissipation of waste heat	8. Use of gas turbines allows waste heat to exhaust to the atmosphere
9. Technology for the disposal of radioactive waste is not established	9. Successful reclamation of stripmine sites is very expensive, and in some cases not demonstrated to be possible

gas cooled reactors (HTGR), conventional coal and oil fired plants, as well as preliminary calculations on coal gasification gas turbine plants (CGGT). Metcalfe's work is used in this report as the source of economic data regarding nuclear power costs.

1.3 Report Outline

In Chapter 2 are developed the model of the coal-gas gas turbine (CGGT) plant used for comparison with the HTGR power station. In this chapter also are outlined the selection of specific components, the sizing of these components and the calculation of fuel consumption rates.

In Chapter 3 are explained the consumer classifications used in the analysis of the thermal and electrical loads of Ft. Bragg. Load schedules for each consumer group are presented. The thermal utility system (TUS) piping distribution system is explained in Chapter 4 together with the design criteria which were used. In Chapter 5 are presented the energy demand simulation results obtained in examining the TUS as described in Chapters 3 and 4. The effect of the consumer thermal-electrical demand mix on TUS loads is also described.

The optimization of the TES with respect to overall cost is discussed in Chapter 6, with Chapter 7 summarizing the report's conclusions and recommendations. Appendices are included to document key technical aspects of the calculations employed to develop the results.

REFERENCES

1. Cochran, N.P., "Oil and Gas from Coal," Scientific American, May, 1976.

2. Metcalfe, L.J., Driscoll, M.J., "Economic Assessment of Nuclear and Fossil-Fired Energy Systems for DOD Installations," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, February, 1975.

CHAPTER 2

COAL GASIFICATION FOSSIL-FIRED GAS TURBINE PLANT ANALYSIS

2.1 Introduction

To ensure a valid economic comparison between a High Temperature Gas Cooled Reactor (HTGR) and a fossil fired alternative, the model of the fossil fired plant should be as well developed and understood as the HTGR model. The fossil-fired plant model should represent realistically the available technology, but not be given credit for potential and as yet undeveloped technological improvements. A Coal Gasification-Gas Turbine (CGGT) plant is selected for analysis based on the preliminary economic comparison performed by Metcalfe. [1] This section of the report outlines the development of the plant model, and describes the final CGGT model.

2.2 Selection of Coal Gasification-Gas Turbine Components

Coal Gasification and Gas Turbine reports [2,3] prepared previously in this project, are used as the basis for the selection of components. The objective of the selection process is the specification of a set of mutually compatible components, well suited to the requirements of a Total Energy Utility System. The selection of a coal gasifier, gas purifier, gas turbine and waste heat exchanger is explained in the following sections.

2.2.1 Gasifier Selection

Table 2.1 (reproduced from the project Coal Gasification Report [2]) summarizes the important system parameters of the currently available commercial coal gasification units. The most crucial of these parameters are those which affect component complexity (and thereby reliability), system compatibility and cost. It is seen that the heating value of the gas should not be considered as a controlling parameter in the selection of process equipment, since relatively simple changes in turbine combustors allow wide variations in fuel heating value. Thus, the greatest weight - in selecting a given component - is given to component compatibility within a complete system, and a history of proven successful performance. Realistically, it should be pointed out that no single gasifier is clearly superior to all others, with the result that the selection of any gasifier would imply gasification costs of approximately the same value.

With these considerations in mind, the Lurgi gasifier is chosen for use in the project's CGGT system because of its history of proven technology, simple construction and reliable operation. Additionally, the output pressure of the Lurgi product gas (300 psi) is suitable for compressed gas storage with minimum compressive work, the Lurgi unit can use air rather than oxygen as a gaseous feedstock (obviating the need for an oxygen plant), and required coal preparation

TABLE 2.1

Competing Advantages and Disadvantages of the Available Gasification Processes

Process	Lurgi	Kellogg McDowell-Wellman	Riley-Stoker	Koppers-Totzek
ADVANTAGES	Under continuing development by the Lurgi Co., and General Electric Co. Produces high pressure gas.	Does not require O_2 feed for high-Btu gas production.	Small unit capacity - permits precise capacity specification.	Rapid startup. Does not consum product gas in steam generati A pressurized p duct gas desigr being developec
DISADVANTAGES	Difficulty with caking coals - Stirring arms required Long successful field experience	Process not improved in recent years, and little current improvement work underway.	Little field experience accumulated.	More extensive coal preparatic required than other processe Needs an oxyger stream.

operations are minimal. It is notable that several other development groups [4,5] have also selected Lurgi gasifiers as the basis for combined cycle plant designs.

The Lurgi does have at least two minor drawbacks (neither of which warrants changing to another gasifier), the low heating value of the product gas, and difficulty in using caking coals. The low heating value of the product gas principally affects the required gas storage volume. The Lurgi Company has treated the caking problem by adding rotating arms, called stirrers, to agitate the coal bed and has successfully gasified caking coals.

2.2.2 Gas Purification

Table 2.2[2] lists a few of the most attractive purification processes available for removing sulfur from the gas. Most proposed large (1000 MWe) [4,5] combined cycle plants use a series of sulfur removal processes, such as potassium carbonate - to Claus purification - to Scott-tails processing. This sequence is used to reduce the loss rate of the catalyst in the Claus purification process by reducing the volume of gas passing through the Claus system. It is thought for the small sized plant proposed for the Ft. Bragg TES (150 MW(t)), that the added cost and complexity of the potassium carbonate system is greater than the corresponding savings in Claus catalyst achieved by using the potassium carbonate system.

TABLE 2.2

H_2S Removal Processes

Process	Sorbent	Final H_2S (ppm)	Operating Temperature (°F)	Final Product	Comments
Purisol	N-methyl-20 pyrrolidone	2	below ambient	H_2S	Expensive sorbent oils and tars mu[illegible] removed; works b[illegible] at high pressure.
Claus	SO_2	100-500	500-600	S	Requires Stretfor[illegible] treatment of tail
Stretford	Na_2CO_3, Na_2VO_5, ADA	1	50-120	S	Expensive sorbent high liquor rates Thiosulfate forma[illegible] minimal below 100 proven process

For this reason, a simple Claus purification system with Stretford tails-processing is recommended for the CGGT plant.

2.2.3 Gas Turbine Selection

Table 2.3 is generated from the project Gas Turbine report [3]. The table displays the principal-characteristics of currently available gas turbines which are relevant to a CGGT plant. The Turbo-Power Marine FT4C Power Pac [3] is selected as the basic unit of electrical generation. Initially, the FT4C was selected for use in the project design because of its unique design which decoupled the electrical generator turbine from the compressor-combustor turbine. This feature would permit a large fraction of the combustion gas flow to by-pass the electrical generator, and to supply heat directly to the Waste Heat Exchanger. It was thought that by-pass flow would be a convenient method of shifting the ratio of electrical/thermal power produced, as the TES demand changed through the day. However, the winter peak thermal load at Fort Bragg is so much greater than the electrical load that merely using FT4C turbines to supply all the thermal power would require additional turbines, with most turbines operating solely as hot water heaters. The solution to this problem is to use a separate gas-fired water heater. Thermal Power (hot water) is produced by the FT4C exhaust waste heat exchangers (as base-loaded heat

TABLE 2.3. RELATIVE MERITS OF THE COMPETING TURBINES EXAMINED IN THIS STUDY

Criterion / System	System Cost	Low BTU Gas Progress	Thermal Control	Fuel Usage	Maintenance Cost	Overall Reliability
TP&M FT4C Aircraft Derivative	-1	0	+2	0	+2	+2
W-501 Westinghouse Industrial Turbine	+1	+1	0	+2	-1	0
W-251 Westinghouse Industrial Turbine	0	+1	0	-1	-1	+2

Key: +2 - Superior
+1 - Good
0 - Fair
-1 - Poor

sources), and also by the gas-fired water heater when necessary.

The question then is, since a gas-fired water heater is being used, why not simply pipe gas to the load points and use conventional heating systems? The answer is made up of two parts.

1) A central station gas-fired water heater (together with the turbine-exhaust water-heaters) reduces fuel consumption and therefore fuel costs. This results in a 91.7% savings in fuel costs (see Appendix A.1),

2) The design concept of the CCGT model is based on a one-for-one replacement of any proposed HTGR/GT plant, powering the Ft. Bragg TES.

For those two reasons, the central station CGGT concept is retained. However, the turbine selection was re-evaluated since the original turbine selection criterion, by-pass flow, was no longer applicable. The FT4C is again selected as the turbine unit of choice because its combustor can be easily modified for use of low BTU gas, its unit size (26.3 Mw(e)) is easily matched to the Ft. Bragg load, and the capital and operating/maintenance costs of the FT4C are reported by utilities [3] as being among the lowest of the available units in the capacity range of interest. It is felt that for increased availability there should be three gas turbine generators, two running and one a backup unit.

2.2.4 Thermal Energy Storage

The thermal load of Ft. Bragg varies typically on a daily cycle as shown by Figure 5.7. There are three ways by which this thermal demand can be supplied:

1. Produce thermal power at the required average daily rate; and use a thermal reservoir to store energy when thermal demand is low, and to release heat when thermal demand is high,
2. Produce thermal energy at the instantaneous rate required by the Thermal Utility System (TUS) load, and
3. some combination of options 1 and 2.

Option 1, thermal energy storage, is the most economical approach for a TES using an HTGR power station, because this option minimizes the size and cost of the HTGR. Since the HTGR is by far the most expensive item in the system, minimizing HTGR cost, as a first approximation minimizes overall system cost.

However, Option 2 could be more attractive for the CGGT system than Option 1. Utilizing Option 2 instead of Option 1 for a CGGT system affects only the designs of the gas fired-water heater, the thermal reservoir, and the gas storage tanks. Implementing Option 2 for a CGGT system requires increasing the size of the gas storage tank(s) so that they can store sufficient gas to permit absorption of the thermal load swings. Option 2 also requires a larger gas fired water heater (sized to meet peak demands), but it eliminates the need for a thermal reservoir. An economic balance must be struck between increasing costs due to increasing gas tank storage volume and water heater size, compared to decreasing costs due to eliminating the thermal reservoir. As is shown in Appendix A.2, it is much less expensive (on a specific energy cost basis) to store energy as hot water than as gas. Therefore, Options 2 and 3 are not considered further in the economic evaluation of possible designs.

Hot water may be stored in steel tanks, pre-stressed concrete vessels, excavated rock caverns or high pressure aquifers. Steel tanks are selected as the storage mechanism, because they have a proven operating history and (for the size range of interest) they may be shop fabricated. Rock cavern or aquifer storage depends on site geology, and since this information was not available (and in any case would vary from site to site) these tehcniques are not considered further.

2.2.5 Gas-Fired Water Heater

Gas-fired water heaters of the required capacity are readily available from several vendors. [6] Two water heaters are used in the CGGT plant to improve system availability. Each gas-fired water heater supplies approximately 35% of the winter peak thermal load, the rest of the thermal energy is recovered from the gas turbine exhaust waste heat exchangers.

2.3 Component Sizing

The size or number of the various components in the CGGT system is set by the loads which these components must serve.

2.3.1 Gas Turbine Sizing

The peak electrical demand of the optimum TES for Ft. Bragg is 50 MW(e). Three TPM FT4C (each 26.3MW(e)) turbine generators are considered to be used to supply this load. Three small units are used (rather than a single larger one) in order to insure a high system availability. Although the FT4C is rated at 26.3 MW(e), it has a reserve capability of 31.1 MW(e) such that in an emergency one FT4C can supply 62% of the peak electrical demand.

Each FT4C has an exhaust waste heat exchanger, which recovers a maximum of 32 MW(t) from the hot exhaust gases. This thermal energy serves the TUS.

2.3.2 Lurgi Gasifier System Sizing

The smallest commercially available gasifier unit has a capacity of 8.00×10^{9} BTU of gas per day. The design winter day requires 1.97×10^{10} BTU of gas, so that three Lurgi units are required. Forced-outage back-up capacity for the Lurgi units could be either a natural gas pipeline to supply gas to the gas turbines and hot water heaters, a combination gas pipeline and electrical grid connection, or tank storage of a suitable liquid fuel such as kerosene. However, three separate gasifiers should be able to maintain reasonably high availability even without back-up capacity.

2.3.3 Sizing the Thermal Reservoir

The largest variation between peak thermal demand and thermal output occurs on the design summer day as shown in Figure 5.35. The energy mis-match between the thermal demand and thermal supply schedules determines the energy storage requirements, and, hence, thermal reservoir size. Integrating the energy schedules mis-match over time (the cross hatched area) shown in Figure 5.35, results in a required energy storage of 509 MW-hr. Using a reservoir water temperature change of from 380 °F to 150 °F, the energy mismatch can be stored in a 123,985 ft^3 reservoir. This corresponds to a tank 54.05 ft in diameter and height. The actual thermal reservoir plant design would probably consist of a set of 6 smaller storage tanks each tank 20 ft. in diameter and 70 ft. long.

2.4 Fuel Consumption

A given space conditioning demand can be supplied by several methods,

1. burning of gas at the load point to supply heat (In Appendix A.1 it is shown that this is very wasteful of energy and money),
2. burning of gas at a central station to produce high temperature water (HTW) to supply TUS loads (more economical than option 1),

3. recovery of thermal energy from the electrical generator turbine exhaust, producing high temperature water (HTW) to supply thermal loads; burning of extra gas as required to meet thermal loads greater than the energy available from waste heat exchangers (more economical than either options 1 or 2)
4. supplying the thermal demand by a combination of electrically-operated heat pumps, HTW heated by turbine exhaust gases, and extra gas burning.

The most economical allocation of electrical space-conditioning and HTW space-conditioning demand is found by determining the thermal loads for various values of electrical/HTW load splits, and then calculating the cost of the corresponding TES. It is found that the total cost of a TES, whether HTGR or CGGT, passes through a minimum at a thermal to electric split of about 75%. Details of fuel consumption and system optimization are explained in Chapter 6. The effect of ambient air temperature variations upon central station efficiency is not considered in these calculations due to the relatively mild climate of the Ft. Bragg area; and thus, the small effect of weather upon plant efficiency.

2.5 <u>CGGT Plant Layout</u>

The size and number of components described in Sections 2.2 through 2.3 are shown in a proposed plant plan in

Fig. 2.1 and a schematic diagram in Fig. 2.2. This layout is not completely optimized, but it does incorporate some features designed to reduce costs and to enhance operational convenience and costs. For example, the gas turbines are located close to the gasifiers and thermal reservoirs. This reduces the gas pipe run from the gasifiers to the turbines, as well as the steam or water lines which run from the waste heat exchangers to the gasifier plant and thermal reservoirs. The gas turbines are arranged so that their exhaust plumes rise in a common area, enhancing overall plume rise.

Because the turbine exhaust waste heat is used to produce hot water for the TUS and is not used in a steam bottoming cycle, there is no need for cooling towers or steam-cycle heat rejection equipment. The plant layout occupies a total of 73,000 ft^2.

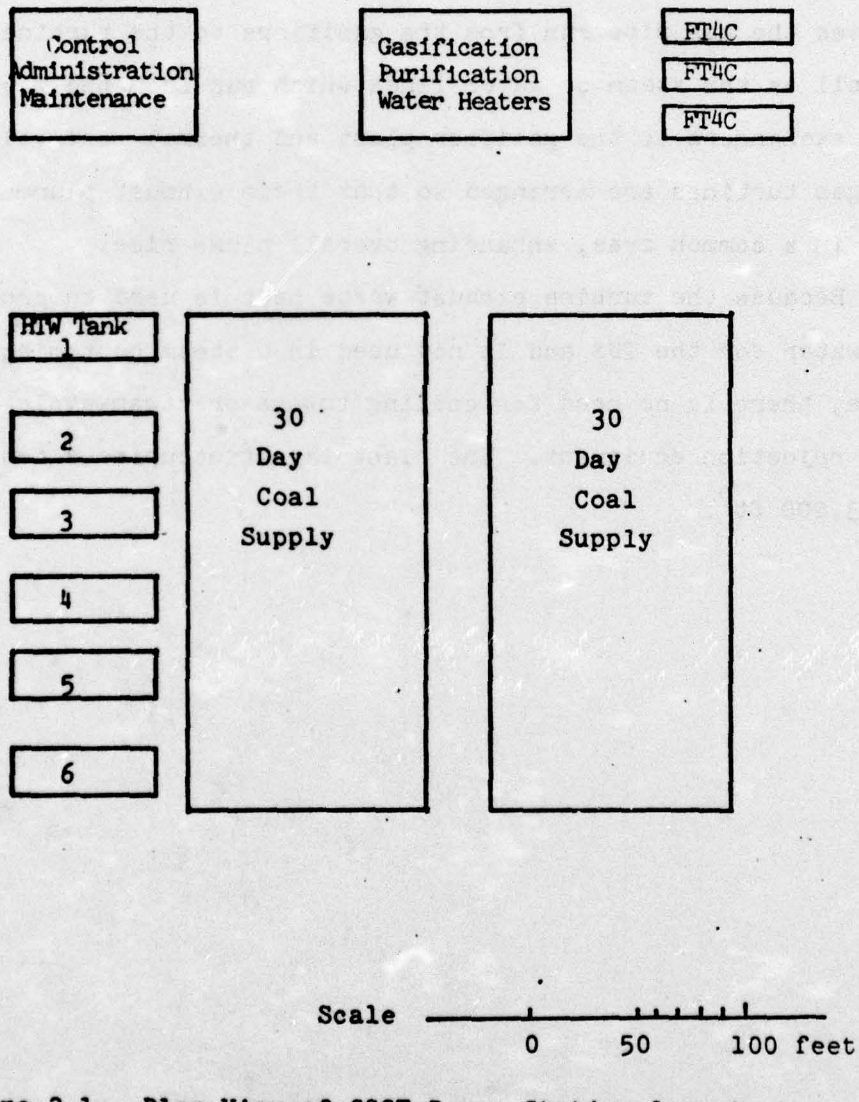


Figure 2.1. Plan View of CGGT Power Station Layout

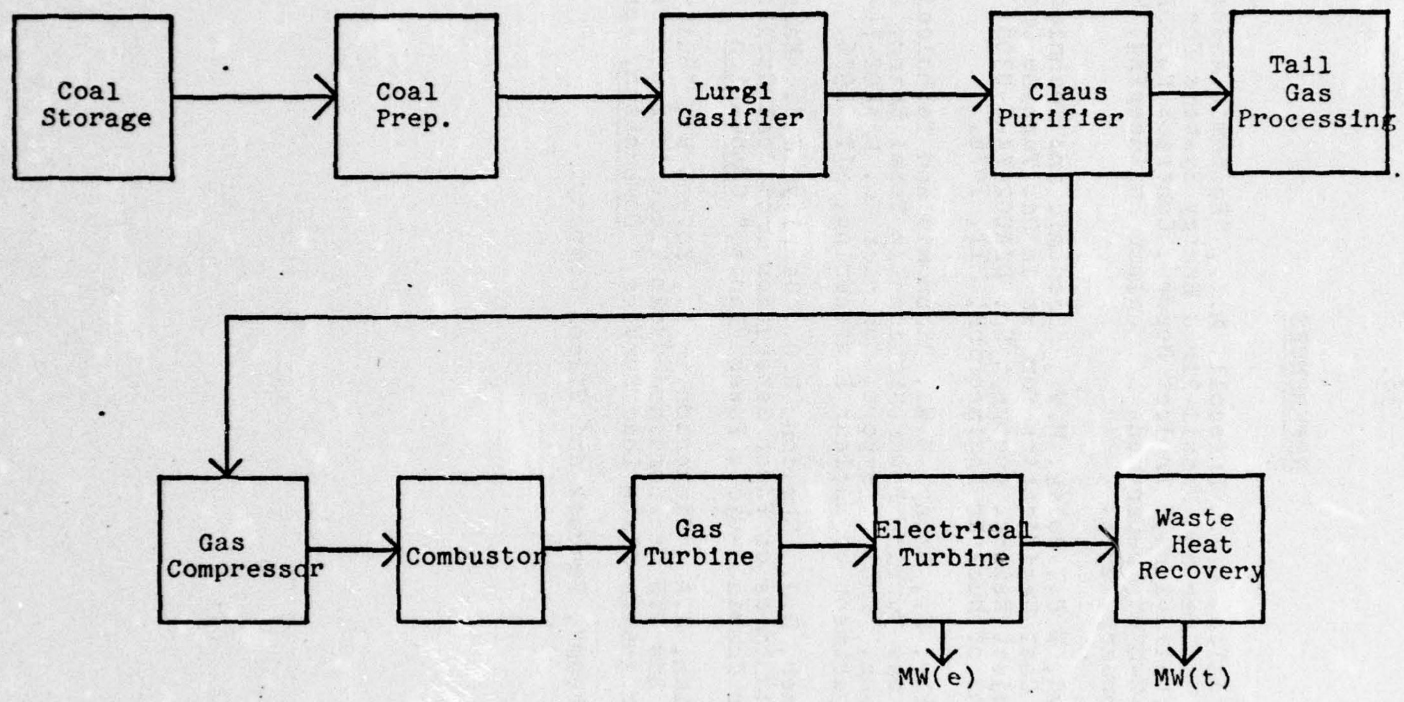


Figure 2.2. Coal Gasification/FFGT Power Plant Schematic Diagram

REFERENCES

1. Metcalfe, L.J., Driscoll, M.J., "Economic Assessment of Nuclear and Fossil-Fired Energy Systems for DOD Installations," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, February, 1975.

2. Boyd, W.C., Golay, M.W., "Economic and Technical Aspects of Coal Gasification for use in Gas Turbine Operation," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, 1976.

3. Kelly, J., Golay, M.W., "Economic and Technical Aspects of Gas Turbine Power Stations in Total Energy Applications," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, 1976.

4. Ahner, D.J., Sheldon, R.C., Garrity, J.J., Kasper, S., "Economics of Power Generation from Coal Gasification For Combined-Cycle Power Plants," Combustion, April, 1976.

5. Alich, J.A., Dickenson, R.L., Koven, N., "Suitability of Low BTU Gas/Combined Cycle Electric Power Generation for Intermediate Load Service," Combustion, April, 1975.

6. "Steam", Babcock and Wilcox Company.

CHAPTER 3

FORT BRAGG ENERGY CONSUMER MODELS

The nominal startup date for the proposed Fort Bragg Total Energy System (TES) is considered to be 1985. To insure that the models of the base's energy consumers accurately reflect anticipated conditions at that future date, the Fort Bragg Master Plan for Future Development has been consulted to identify the building types and base configuration to be used in the system analysis. Following extensive discussions with personnel at the U.S. Army Facilities Engineering Support Agency (FESA) at Fort Belvoir, Virginia, it has been concluded that the buildings at Fort Bragg may be aggregated into a total of eleven general energy consumption categories based upon documented building usage and construction characteristics. Table 3.1 lists these eleven classes with brief descriptions of the "typical" units chosen to represent each category and the number of each found on the base. Appendix B contains more complete descriptions of these buildings, including their construction and usage specifications supplied as input data to the TDIST consumer modelling subroutines.

The magnitudes of the total energy demands of these consumers on the peak winter heating and peak summer cooling days determine the design criteria to be met by the compo-

TABLE 3.1

FORT BRAGG BUILDING CATEGORY DESCRIPTIONS

1. Troop Housing Modern: These new and planned troop barracks are composed of three-story concrete and stucco modules, each designed to house approximately 60 enlisted personnel; three or four residence modules are typically combined with a building service module to form a barracks unit. For modelling purposes, a single residence module is considered to be the representative unit, having a total floor area of 6172 ft^2. A total of 159 modules are considered to be located at Fort Bragg in 1985.

2. Family Housing Modern: 3648: Modern family housing includes all housing units dating back to roughly the 1950's and, for convenience in modelling, this category has been divided into two sub-categories: two-family dwellings are included in the 3648 class and four-family dwellings are designated as row houses as described below. The representative single-story two-family unit is of brick construction and has a total floor area of 3648 ft^2. A total of 1844 of these units are on the 1985 base.

TABLE 3.1 (continued)

3. Family Housing Modern: Row: The four-family modern housing units consist of a mixture of two-story brick, and combined brick and frame construction units which, in some cases, are physically attached to form larger connected housing groups. The total floor area of a typical unit is 7500 ft^2; 359 units are distributed throughout the base.

4. Family Housing: 34: This category of family housing consists primarily of large brick single-family residences dating back to the 1930's. At Ft. Bragg, these units are principally used by high ranking officers and tend to represent the best available accomodations. A representative floor area is taken to be 4147 ft^2, and a total of 115 units exist on the base.

5. Fort Bragg Hospital: Since the hospital is a unique building, and since it represents approximately 1% of the total base load, a separate building class is allocated to it. The building itself is composed of several sections reflecting many additions over the years. It normally contains approximately 500 beds in a total floor area of 411,053 ft^2.

TABLE 3.1 (continued)

6. Storage: Although many unrelated storage facilities exist at Fort Bragg, it was decided to combine them all into a single class due to their similarity of use and their relatively small contributions to the base's total load. Construction and sizes of these buildings vary considerably, but the representative unit was chosen to be a large warehouse with a floor area of 11,421 ft^2. A total of 26 of these units are specified.

7. Community: Perhaps the widest range of diverse building constructions and usage patterns is included in this class. Facilities range from recreation buildings to retail sales establishments, units which individually contribute little to the base demand but which in total represent a significant load. The representative unit is assumed to have a floor area of 20,486 ft^2, and 41 of these buildings are located throughout the base.

8. Administration and Training: The age and construction of these buildings varies considerably from unit to unit, with the typical structure being formed of a reinforced concrete foundation, brick walls, and a

TABLE 3.1 (continued)

8. Administration and Training (continued)
built-up roof. The representative unit is three stories tall with a total floor area of 24,114 ft^2. 56 of these buildings will exist at Fort Bragg in 1985.

9. Operations and Maintenance: A machine shop has been chosen to be representative of a wide variety of maintenance buildings on the base. General construction includes either block and steel or brick and block walls, a reinforced concrete foundation, and a built-up roof. The average floor area is 41,850 ft^2, and 29 of these buildings have been identified.

10. Troop Housing: Brick: These barracks units are relatively modern three-story dwellings with a capacity of roughly 200 men each. Construction is of brick, and a representative unit has a floor area of 50,959 ft^2. A total of 26 of these units will exist on the 1985 base.

11. Troop Housing: Block: Similar in size to the brick units described above, the block barracks consist largely of older renovated units having an average capacity of 160 men each. Construction is of

TABLE 3.1 (continued)

11. Troop Housing (continued)
reinforced concrete and blocks with an average floor area of 50,959 ft^2. 52 units are identified on the base.

nents of the thermal utility system and, depending upon how these demands are supplied, set the required power plant installed capacity and its rated thermal-to-electrical energy output ratio. Similarly, the variations in the thermal loads on these days dictate the installed system thermal energy storage capacity required to smooth the imbalances between the diurnal thermal and electrical energy demand schedules. The choice of these design days is thus critical to the ultimate design, configuration and cost of the TES; the weather conditions must be severe enough to insure that the system is capable of meeting the maximum annual power demands, but they must not be so extreme as to cause the system to be grossly over-designed and much more costly than necessary. Following a fairly extensive analysis of recent historical weather data for Fort Bragg, it has been decided to use for the design days the summer and winter hourly air temperatures shown in Table 3.2. Although records from the National Oceanic and Atmospheric Administration's weather station at Fayettville (located approximately five miles to the southeast of Fort Bragg's eastern boundary) indicate that the extreme temperatures for this location during the past 40 years range from 5 °F to 102 °F, the minimum and maximum values of 15 °F and 95 °F shown in Table 3.2 were chosen as being representative of conditions occurring with a relatively high annual

TABLE 3.2

DESIGN DAY AIR TEMPERATURES

Time		Winter Day, °F	Summer Day, °F
12		22	83
1		21	80
2		18	79
3		15	78
4	AM	16	78
5		17	79
6		18	80
7		21	81
8		23	82
9		26	86
10		27	90
11		30	92
12		33	93
1		35	94
2		33	95
3		32	95
4	PM	29	94
5		30	92
6		30	90
7		30	89
8		29	88
9		27	87
10		25	86
11		23	85
12		22	83

expectancy. Because only daily maximum and minimum temperatures were readily available for the Fayettville weather station, these extremes have been used in combination with typical winter and summer day temperature schedules for Boston to generate the given diurnal variations. In designing the final system, the specified extremes have been broadened somewhat to 10 °F minimum winter air temperature and 100 °F maximum summer air temperature for conservative sizing of the TES components in single period, steady-state calculations.

For simplicity, and because coincident wind velocity data was not readily available during the system design period, a constant wind velocity of 15 mph from the west has been assumed throughout both design days. The nominal peak solar radiation intensity at Fort Bragg for the winter day was assumed to be 390 BTU/hr per square foot of horizontal surface area; the summer day peak was 344 BTU/hr per square foot.[1] Cloudless skies have been assumed, but normal seasonal atmospheric haze and diffusion effects are included as modifying these direct solar radiation intensities. Summer day building usage and occupancy characteristics have been shifted in time by one hour to account for the effects of Daylight Savings Time, but all load calculations and results are presented in real solar time to allow direct comparison among load profiles at different times of the year.

Figures 3.1-3.11 present the design day space conditioning energy demand schedules computed for each of the eleven consumer categories. It should be noted that these schedules represent only the net hourly energy gains or losses from the buildings. The corresponding demands to be met by the thermal and electrical energy distribution networks will, of course, depend upon the types and efficiency of the space conditioning equipment used to supply these requirements.

The shapes of the load schedules illustrate the relative effects of the major components of the space conditioning demands. The winter minimum and summer maximum occurring during the daylight hours are due principally to solar radiational heating. These solar effects are compounded, especially for the commercial and public-use building categories, by heat generated internally from lighting and equipment usage. (In fact, for the Administration and Training class, Fig. 3.8, the combined effects of solar and internal heating between noon and 1 P.M. on the winter day reduce that building's heating load to zero, even though the outside air temperature is only 33° to 35 °F. Consultation with Army personnel at FESA and an independent analysis by Michael Baker, Jr. of New York, Inc. [2] have verified this behavior for these particular buildings.)

All forced air ventilation and induced infiltration air flows are assumed to be direct air exchanges between the interior and exterior of the buildings. The significant effects of these components of the space conditioning loads are evident in the winter demand schedules for the hospital (Fig. 3.5), the ventilation requirements of which are large and the usage of which is fairly constant throughout the day, and for the community buildings (Fig. 3.7), the afternoon and evening usage and large ventilation requirements of which during occupancy cause both its summer and winter demand curves to be skewed slightly more toward the evening hours than those of the other building types. (The winter day profile for the hospital, while irregular, is relatively flat compared with those of the other categories due to the hospital's fairly uniform occupancy characteristics and the offsetting effects of slightly higher ventilation requirements and solar heating during the day; the large variation in its summer day demand occurs due to the additive effects of these components when the ambient air is at a higher temperature than that desired within the building). The Storage (Fig. 3.6) and Operations (Fig. 3.9) building categories are assumed to have no air conditioning (see Appendix B). Since this condition is transmitted to the TDIST consumer demand models only by requiring the internal room temperatures to vary directly with the outside air temperature, during periods of sunlight the combined effects of solar heating and a small amount of internal lighting produce the nominal summer day cooling demands shown. As has been mentioned previously, these demands are applied to the energy supply systems only through the use of specified space conditioning equipment units. Since no air conditioning is desired for these two categories, their cooling loads do not appear on the system, and the calculated positive energy demands merely indicate that the actual building temperatures are somewhat higher than the outside air temperature. The demand profiles for all the Troop Housing and Family Housing categories exhibit the same qualititative behavior, reflecting the general similarity of occupancy of these units during the late afternoon and evening hours and the dominance of solar heating during the day.

Figure 3.1

Design Space Conditioning Demands
Type 1: Troop Housing Modern
Unit: Single Residence Module

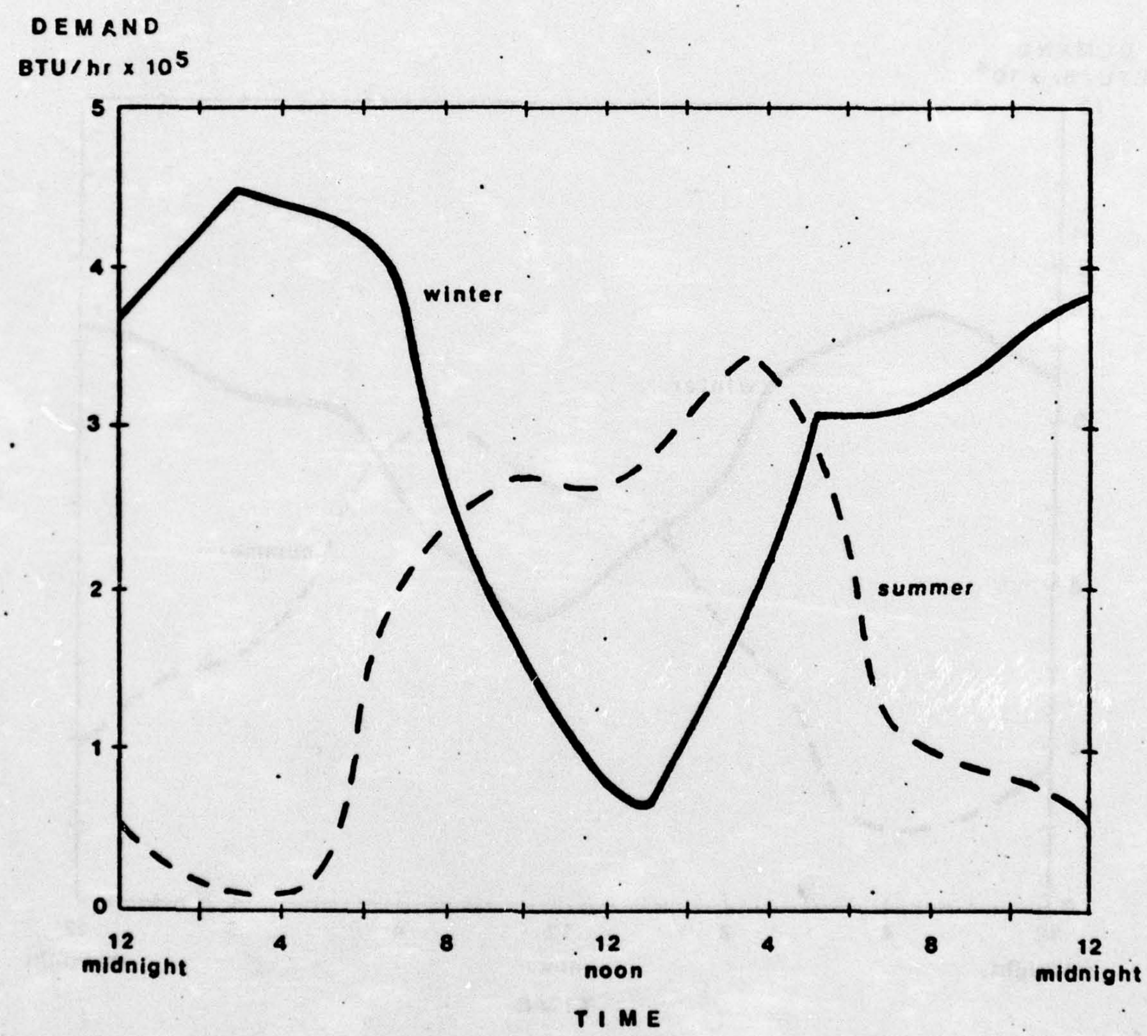


Figure 3.2

Design Space Conditioning Demands
Type 2: Family Housinh Modern: 3648
Unit: Two Family Duplex

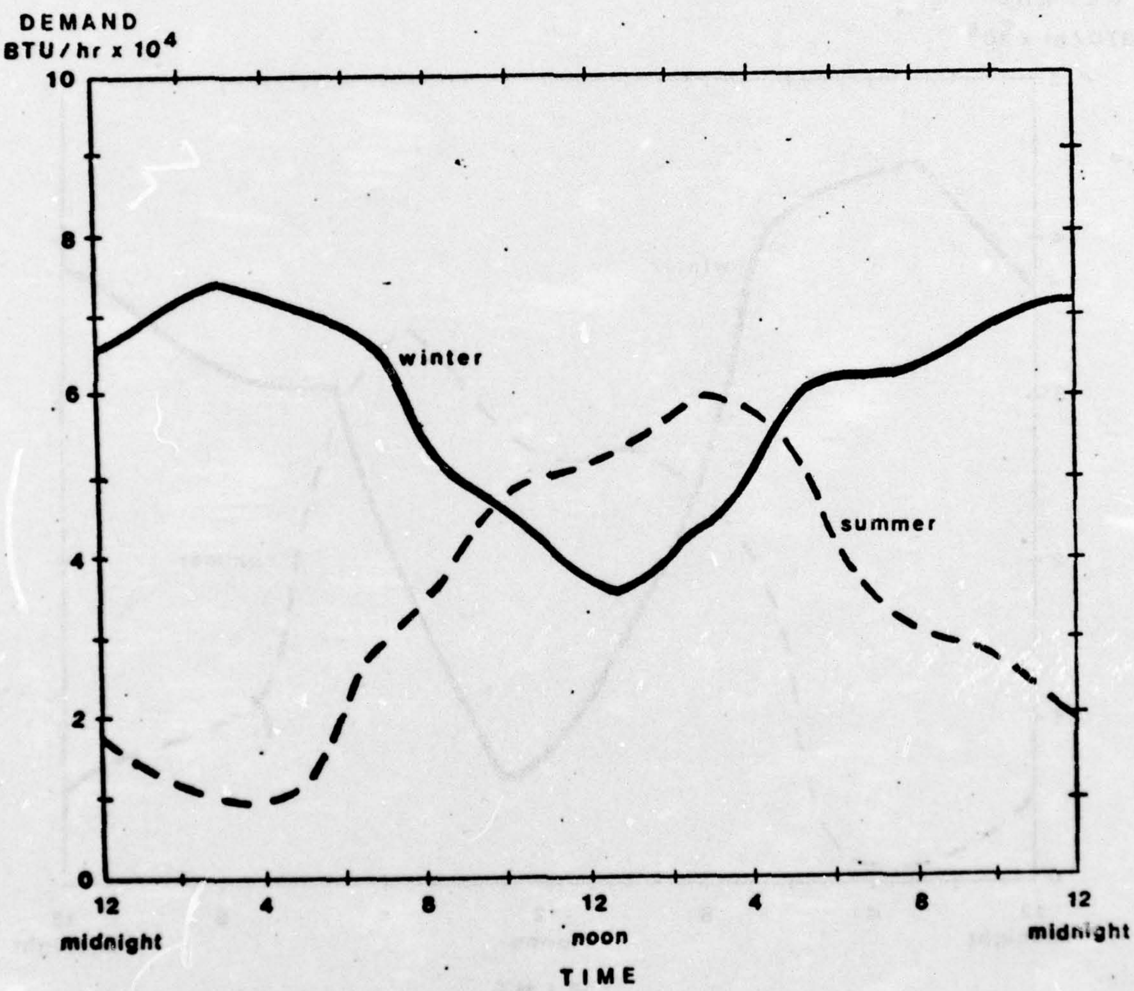


Figure 3.3

Design Space Conditioning Demands
Type 3: Family Housing Modern: Row
Unit: Four Family Dwelling

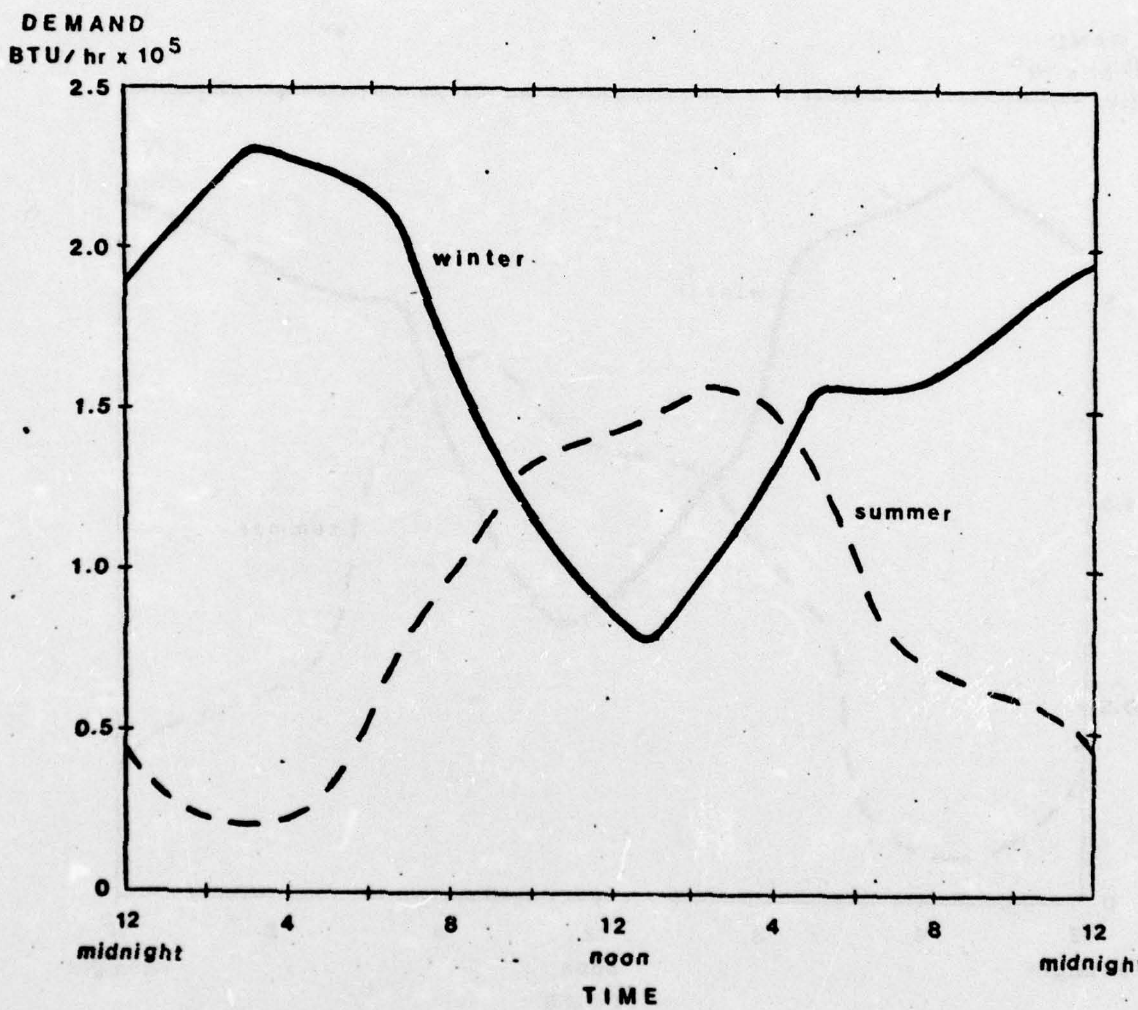


Figure 3.4

Design Space Conditioning Demands
Type 4: Family Housing: 34
Unit: Single Family Detached Dwelling

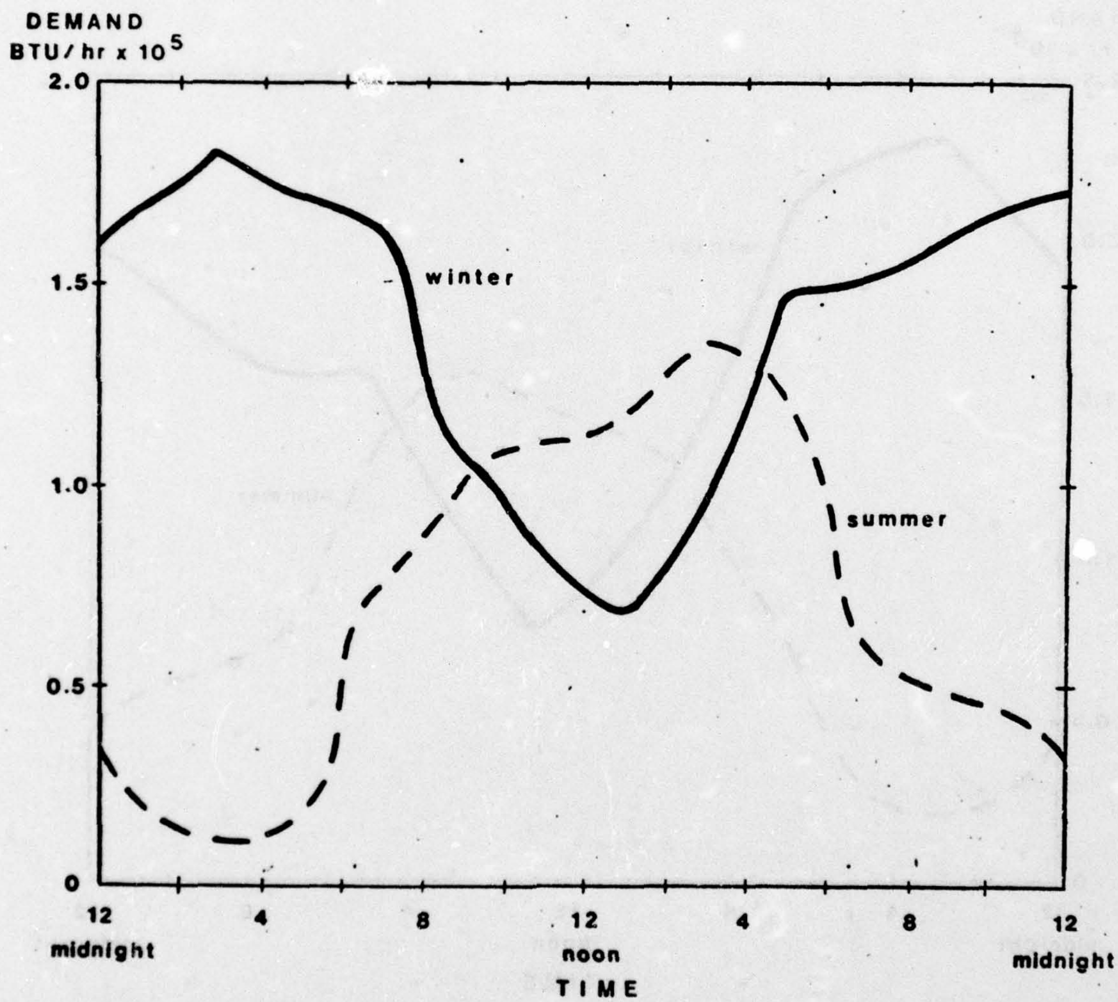


Figure 3.5

Design Space Conditioning Demands
Type 5: Fort Bragg Hospital
Unit: Hospital

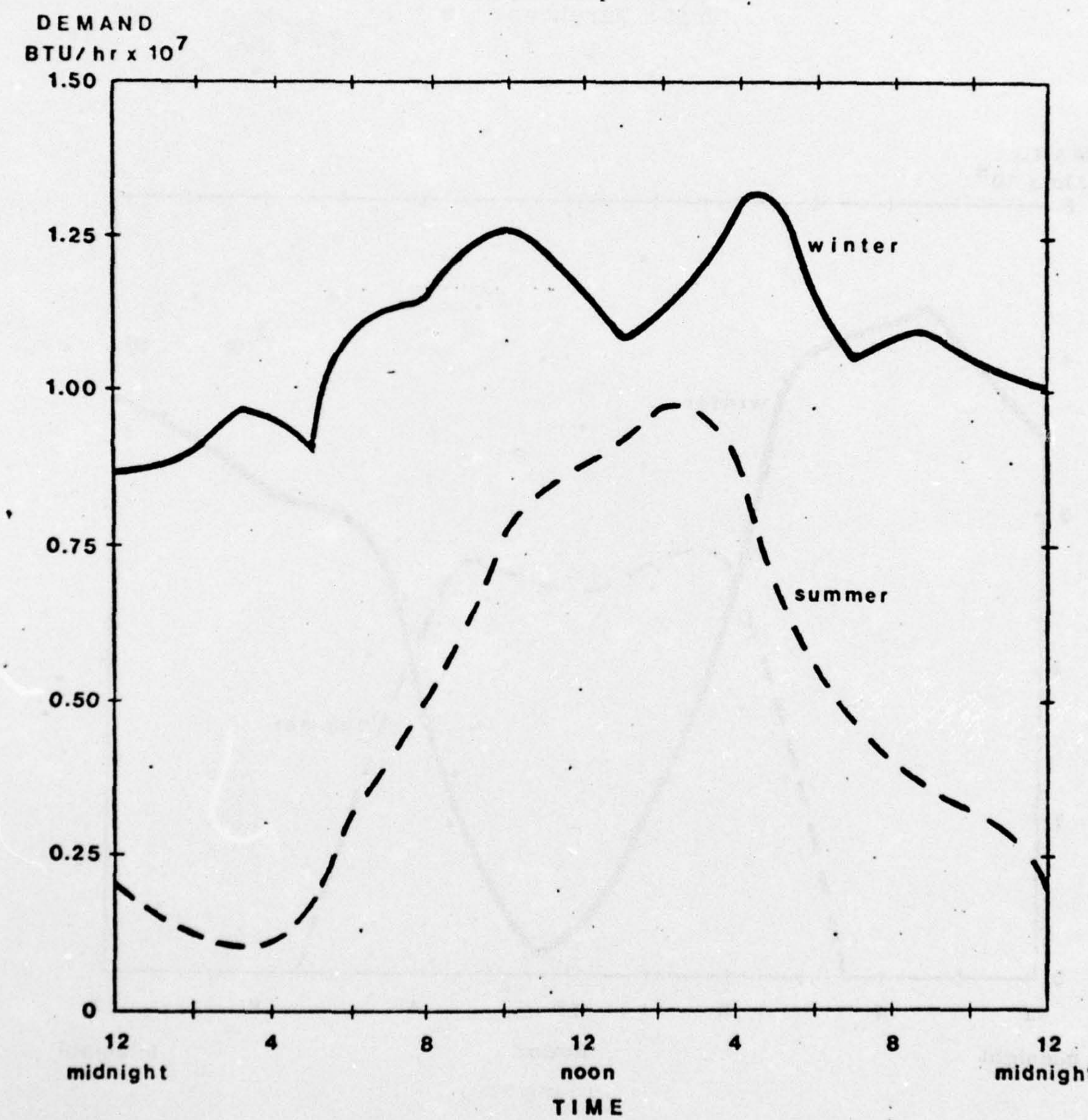


Figure 3.6
Design Space Conditioning Demands
Type 6: Storage
Unit: Warehouse

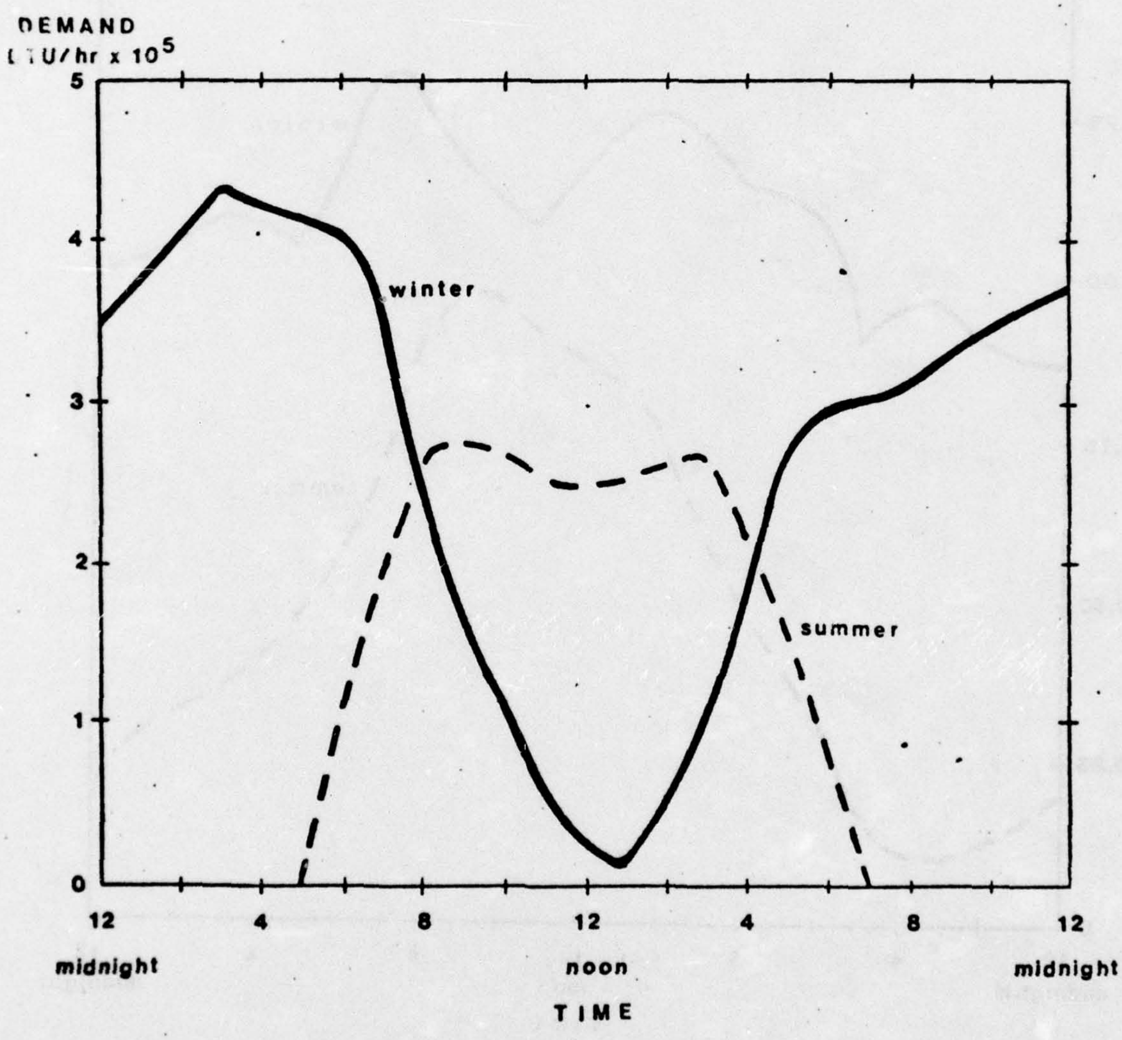


Figure 3.7

Design Space Conditioning Demands
Type 7: Community
Unit: Recreation/Community Center

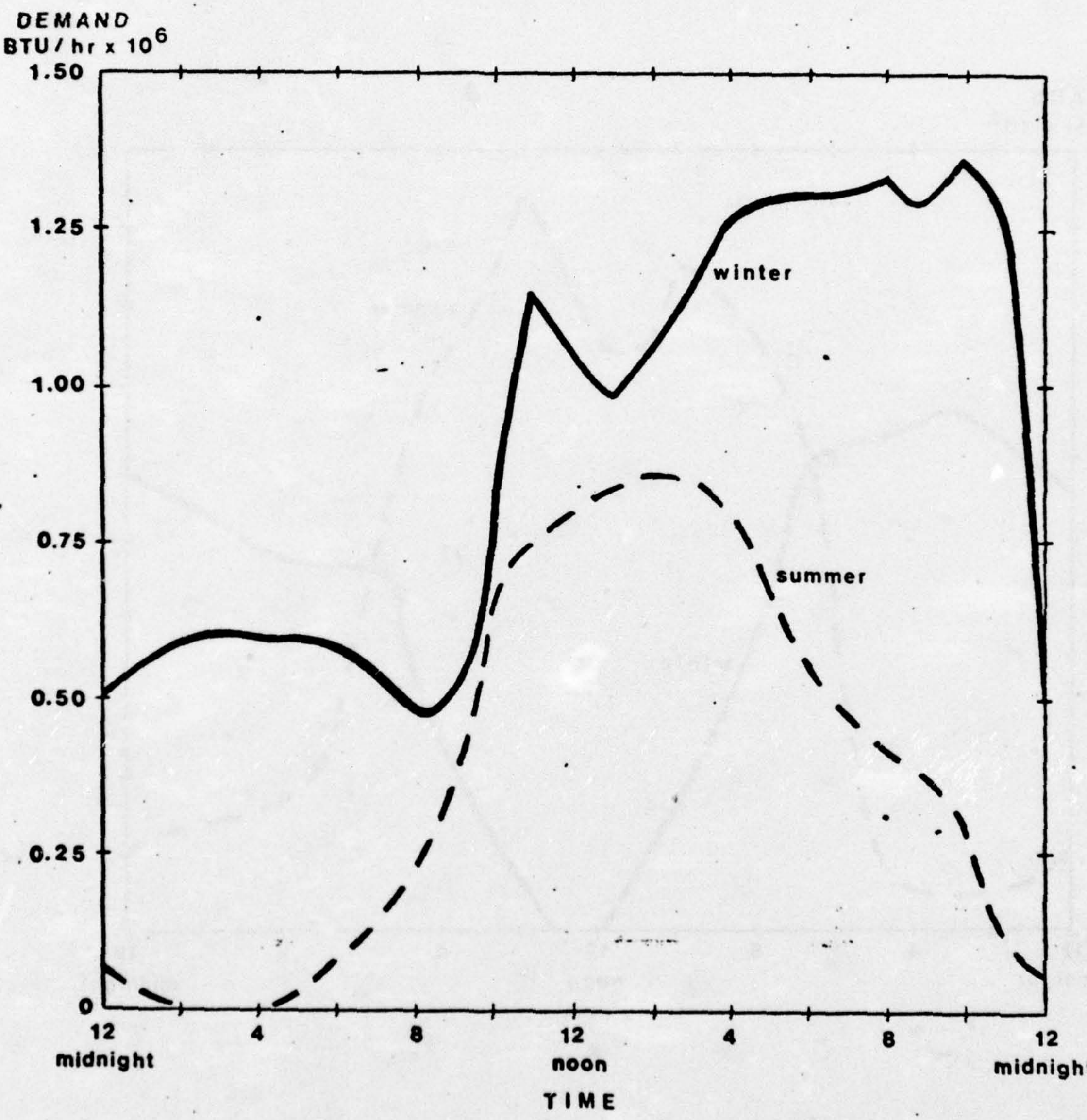


Figure 3.8

Design Space Conditioning Demands
Type 8: Administration and Training
Unit: Training Building

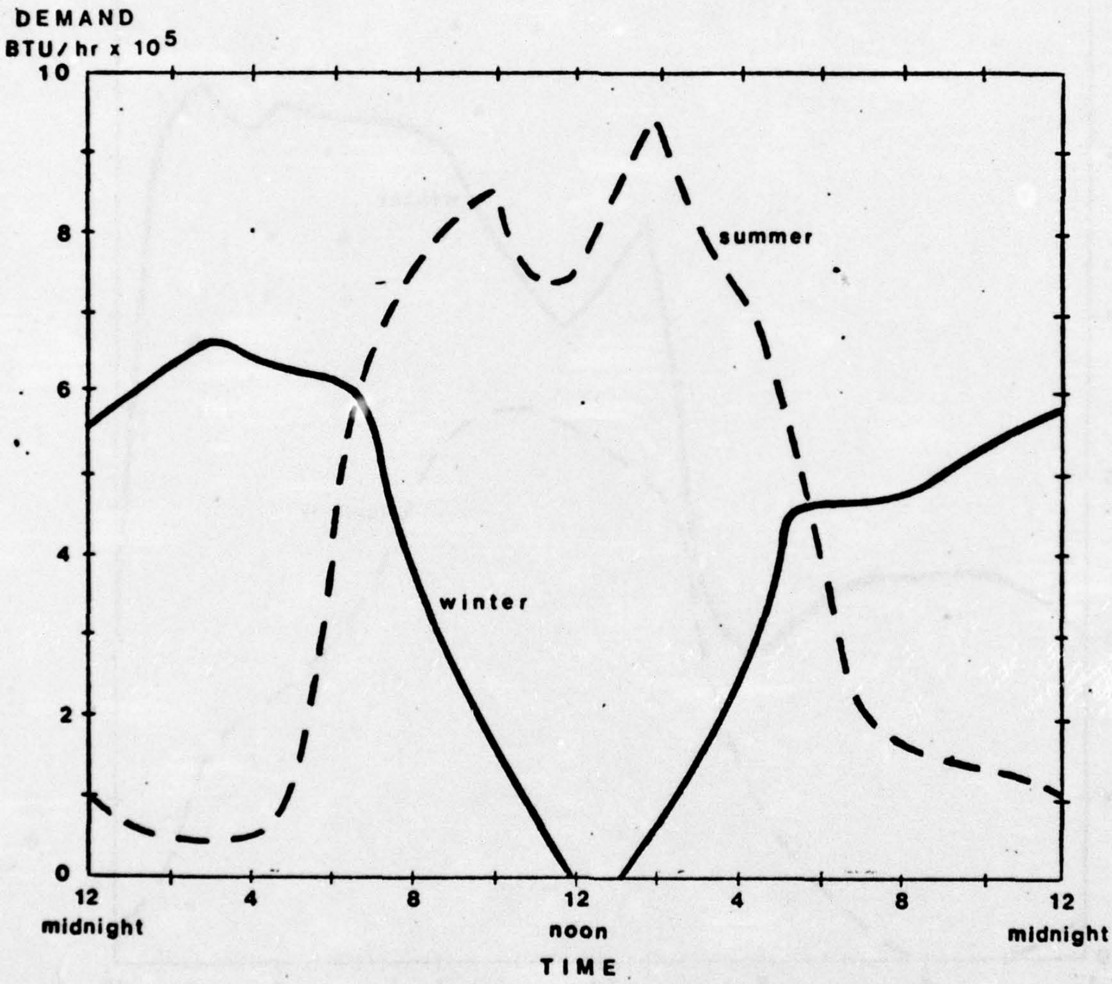


Figure 3.9

Design Space Conditioning Demands
Type 9: Operations and Maintenance
Unit: Machine Shop

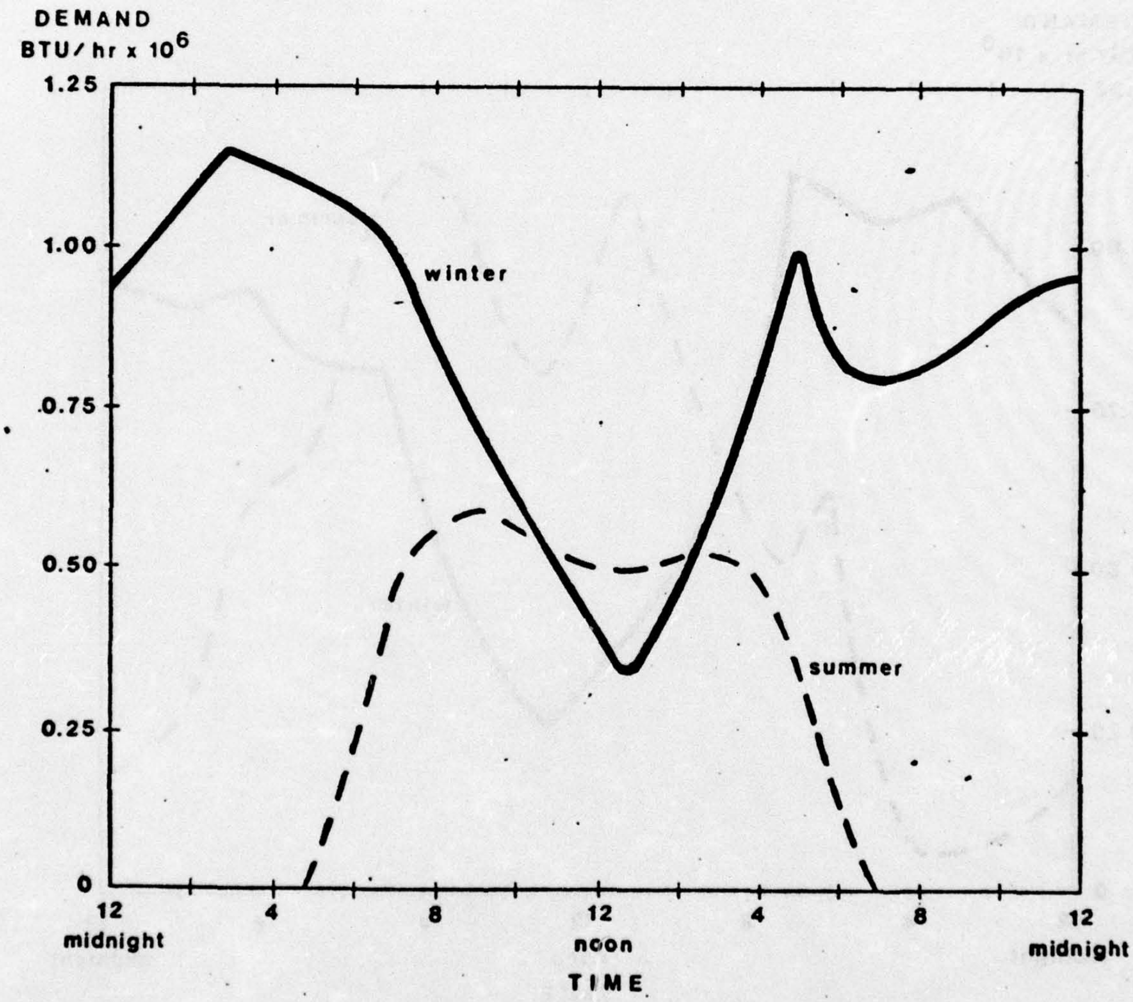


Figure 3.10

Design Space Conditioning Demands
Type 10: Troop Housing: Brick
Unit: Barracks Unit

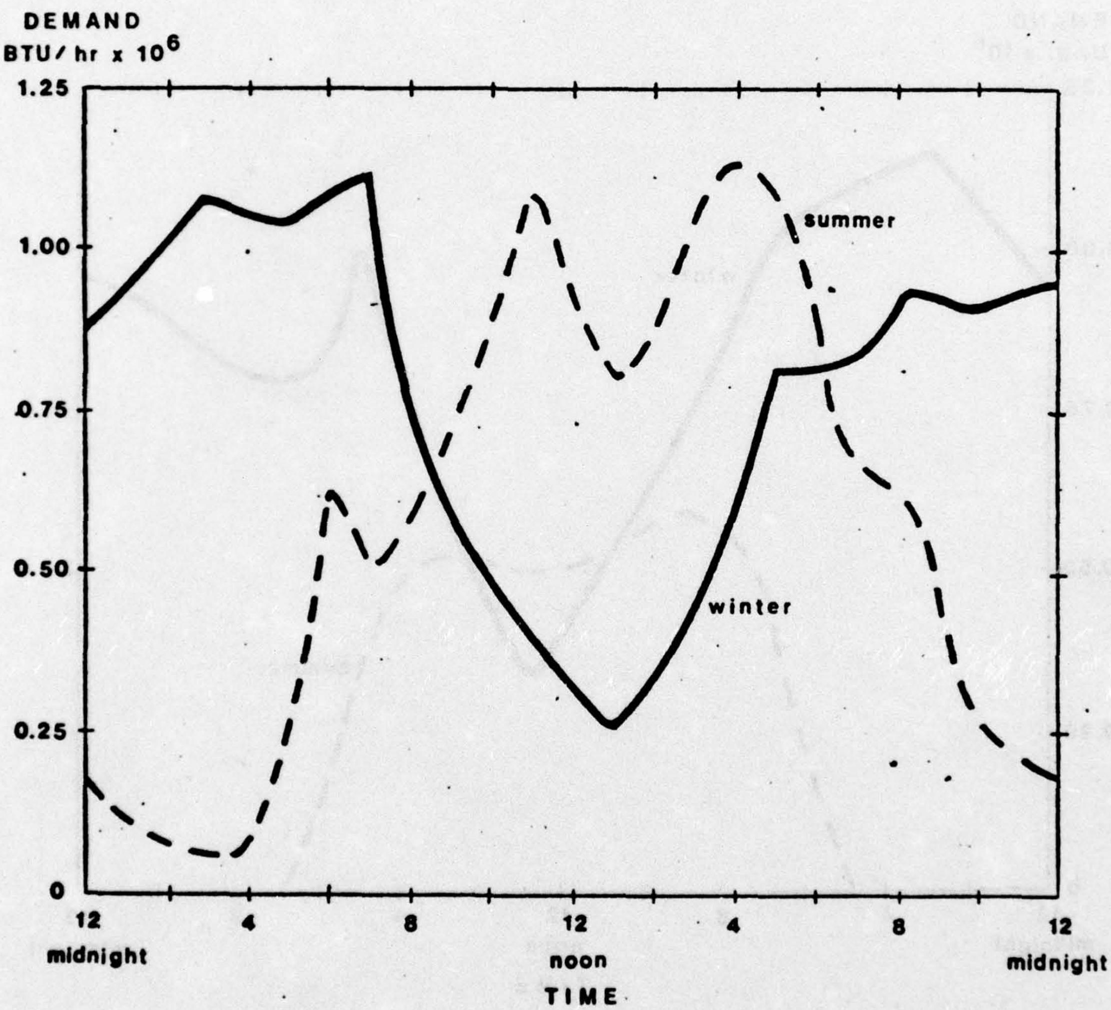
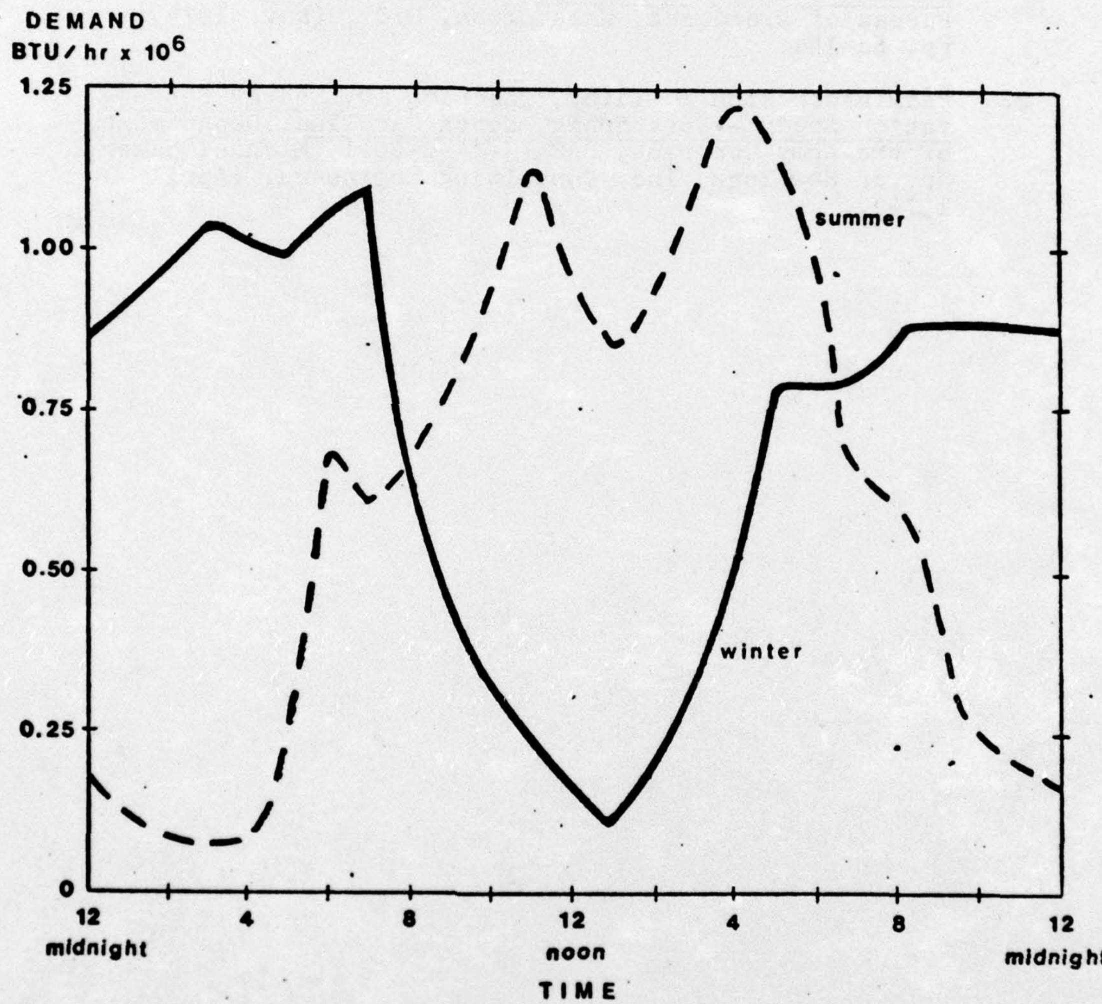


Figure 3.11

Design Space Conditioning Demands
Type 11: Troop Housing: Block
Unit: Barracks Unit



References

1. Kusuda, T., "SUN, An Algorithm to Find Solar Position, and Intensity of Direct Normal and Diffuse Radiation," NBSLD, Computer Program for Heating and Cooling Loads in Buildings, NBSIR-74-574, National Bureau of Standards, Washington, D.C., (Nov. 1974), pp. 6a-14a.

2. "Administration Building," Section 10, Energy Conservation Study - Fort Bragg, North Carolina, Department of the Army Contract, DACA73-74-C-0011, Michael Baker, Jr. of New York, Inc. Consulting Engineers, (April 1974).

CHAPTER 4

FORT BRAGG THERMAL UTILITY SYSTEM OPTIONS

Encompassing an area of approximately 17 square miles, the inhabited section of Fort Bragg occupies the extreme eastern end of the base. (The remainder of the base, an area of roughly 187 square miles, is used for training grounds, firing ranges, etc., and has very few permanent buildings.) The population of Fort Bragg in 1985 is expected to be approximately 44,000, divided in a ratio of roughly 53% single enlisted personnel and 47% resident families. Supplying these residents with both thermal and electrical energy from a single power plant requires carefully designed piping systems and electrical distribution circuits to deliver the necessary energy at a minimum total cost. Since virtually every building on the base will require some form of electrical service regardless of the design or configuration of the proposed total utility system, it has been assumed that the components and costs of the electrical distribution network will be determined relatively independently of the final system choice and will be incurred whether or not the TES proposal is adopted. Therefore in examining a range of utility system options for Fort Bragg, primary emphasis has been placed upon investigating the technical and economic variations in the thermal energy distribution network, the power plant, the thermal energy storage reservoir and any necessary auxiliary power plant cooling systems, with secondary effects upon the electrical network being noted where they are deemed important.

Figure 4.1 is a planning map for Fort Bragg illustrating the layout of the inhabited area of the base as it is expected to appear in 1985. (A majority of the buildings shown exist today, with the addition of the two residential developments at the base's southern extremity and the replacement of World War II vintage temporary buildings with modern troop housing complexes being the major changes planned during the next decade).

One of the ground rules established early in the Fort Bragg study was that the proposed TES be nominally capable of supplying the base's total annual energy demands without relying upon any auxiliary capacity from outside the Fort's boundaries. Because some loads (space conditioning, domestic hot water) readily lend themselves to either thermal or electrical energy supplies, the possibility arises for optimizing the TES to obtain the highest average efficiency and lowest total cost through carefully designed tradeoffs between the percentages of the consumers utilizing the power plant's thermal and electrical energy outputs. Ideally, the optimum system design would be that configuration

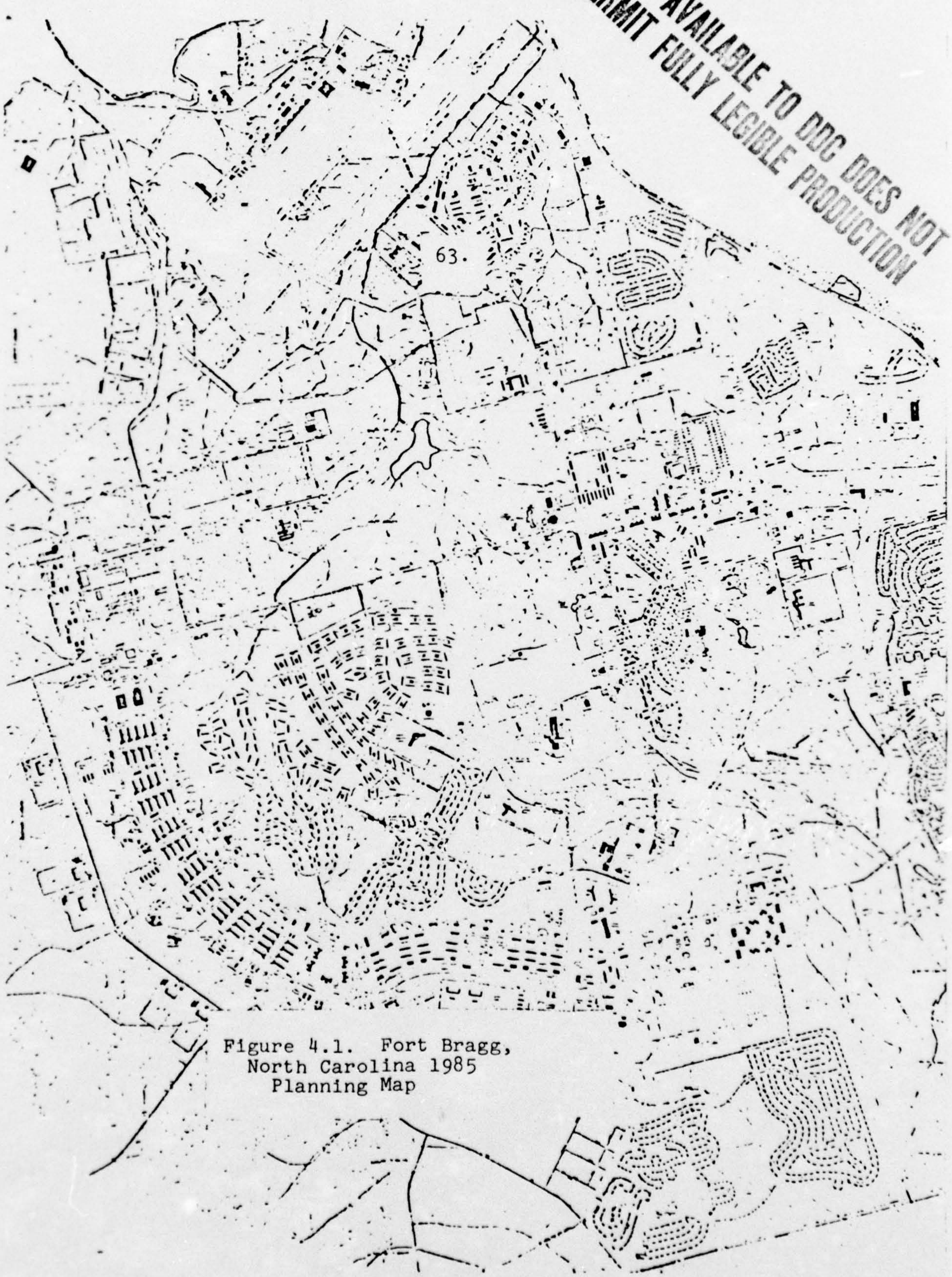


Figure 4.1. Fort Bragg, North Carolina 1985 Planning Map

which, throughout the entire year, would cause electrical and thermal energy to be produced and consumed in a ratio such that none of the power plant's total output would be wasted. (Chapter 6 more fully discusses this optimization problem and its practical design limitations). As demand variables for the TES design process, three general load categories are specified: space conditioning served thermally or electrically, domestic hot water service supplied thermally or electrically, and non-space-conditioning electrical demands (motors, appliances, lighting, etc.). The space conditioning and domestic hot water demands are computed on an individual building unit basis to allow their supply modes and service equipment to be varied. The non-space-conditioning electrical loads, because of a lack of individual unit consumption data and the existence of municipal service components impossible to associate with any single energy consumers, are aggregated into a single electrical demand schedule for the entire base, which was obtained from central metering equipment at Fort Bragg.

4.1 100% Thermal Supply Option

4.1.1 Primary and Secondary Loop Piping

The first thermal utility system option studied for the base assumes that every building's space conditioning and domestic hot water energy demands are supplied by

Figure 4.2. "100% Thermal" Case
Utility System Piping Layout


N
1
2
3
5
7
8
9
10
12
13
15
16
17
FORT BRAGG
MASTER PLAN
FUTURE DEVELOPMENT PLAN
BUILDING AREA PL.. AN

#1-#3 at the center of the base and returning through the same trench to the heat exchanger for loop #4 and thence to the power plant, which is assumed to be located at Bones Ford, the junction of two small streams in a valley approximately 1-1/2 miles to the west of the western end of the primary loop shown in the figure. Similarly, each of the dashed secondary loop lines represents both supply and return pipes for each load center heat exchanger, which join at common supply and return headers to form a single flowstream through the loop's supply heat exchanger.

The load center heat exchangers for the 100% Thermal case are used in the TDIST simulation models as a convenient set of indices for identifying the locations of the buildings served by the thermal utility system. Table 4.1 presents the distribution of the eleven specified buildings types described in Chapter 3, indexed according to the load center heat exchanger numbering scheme shown in Fig. 4.2. Table 4.2 shows the total pipe lengths, nominal initial diameters and insulation parameters for each of the five loops shown in Fig. 4.2. (The final optimum system design pipe diameters are set after considering the effects of frictional pressure losses around each of the loops as discussed in Chapter 6.)

TABLE 4.1

FORT BRAGG BUILDING DISTRIBUTION

Load Center Heat Exchanger (see Fig. 4.2)	Building Type (see Table 3.1) 1	2	3	4	5	6	7	8	9	10	11
5	75							12	2		
6							13	10	3	10	
7		252						1			
8		298									
9			76				1	1			
10		363	26								
11		370	41								
12	75							9	3		
13	9	30		18	1		6	1			
14				43				3	11		
15						13		9			7
16				54				4			
17		292						1			
18		68	50								
19							12	2	9		28
20						13	8	1	1	16	17
21		171					1	1			
22			166					1			
TOTAL	159	1844	359	115	1	26	41	56	29	26	52

TABLE 4.2

FORT BRAGG 100% THERMAL CASE PIPE PARAMETERS

Loop (1)	Total Length	Pipe O.D. (2)	Insulation (3)
1	6.3 miles	12"	Type 1 (4)
2	7.8 miles	12"	Type 1 (5)
3	7.5 miles	12"	Type 1 (6)
4	6.2 miles	12"	Type 1 (7)
Primary	10.0 miles	18"	Type 2 (8)

(1) See Fig. 4.2

(2) Nominal initial simulation diameters. Final values for optimum system set according to frictional losses as discussed in Chapter 6

(3) Type 1 insulation (12" pipe): 3" calcium silicate insulation, 1" air space, 10 gauge galvanized steel spiral welded conduit, asphalt impregnated fiberglass screen, fiberglass reinforced asbestos pipe line felt, buried 6 ft. deep on center. (Meets Army Corps of Engineers specification CE-301.21)

Type 2 insulation (18" pipe): 4" calcium silicate insulation, 1" air space, 10 gauge galvanized steel spiral welded conduit, asphalt impregnated fiberglass screen, fiberglass reinforced asbestos pipe line felt, buried 6 ft. deep on center. (Meets Army Corps of Engineers specification CE-301.21)

(4) Return piping uninsulated, approximately 3.1 miles

(5) Return piping uninsulated, approximately 3.9 miles

(6) Return piping uninsulated, approximately 3.7 miles

(7) Return piping uninsulated, approximately 3.1 miles

(8) Return piping from secondary heat exchanger #4 to power plant uninsulated, approximately 3.1 miles

4.1.2 Tertiary Piping

Although not explicitly included in any of the TDIST simulations due to its assumed small effects upon the overall system thermal inertia and control stability, the extensive system of "tertiary" piping connecting the individual energy consumers with their respective load center heat exchangers represents a significant contribution to the total costs of the thermal utility system and must be included in any economic analyses of the TES. Table 4.3 presents the total tertiary piping lengths associated with each of the 18 load centers in the 100% Thermal case. Although no detailed optimization calculations have been performed for this tertiary piping network, the mains are sized such that the maximum fluid velocity in each distribution loop does not exceed 10 feet per second under design load conditions. Table 4.4 shows the nominal pipe sizes chosen for the three levels of the tertiary system at each load center heat exchanger. Supply piping is assumed to be insulated; return piping is assumed to have no insulation. All pipes are assumed to be buried double in 6-foot deep trenches.

4.1.3 Heat Exchangers

In the 100% Thermal supply case, all buildings are assumed to be heated by hot water from either in-house heat exchangers or directly from the thermal utility system

TABLE 4.3

100% THERMAL CASE TERTIARY PIPING

Load Center Heat Exchanger (see Fig. 4.2)	Tertiary Piping, miles Main(1)	Branch(2)	Service(3)	Total
5	2.1	1.9	2.2	6.2
6	2.0	3.2	3.2	8.4
7	4.8	2.3	7.3	14.4
8	6.9	1.1	7.6	15.6
9	2.4	0.6	6.3	9.3
10	5.8	0.5	9.8	16.1
11	5.2	3.0	10.4	18.6
12	0.3	1.3	2.1	3.7
13	2.7	0.9	3.9	7.5
14	4.4	2.0	5.1	11.5
15	2.1	1.3	2.9	6.3
16	2.3	0.4	2.3	5.0
17	5.6	5.2	7.5	18.3
18	3.2	2.7	4.9	10.8
19	3.0	3.1	5.6	11.7
20	2.3	4.2	4.6	11.1
21	4.2	2.3	5.0	11.5
22	2.9	3.1	7.5	13.5
TOTAL	62.2	39.1	98.2	199.5

(1) Main piping is the principal distribution circuits running under major streets from the load center heat exchangers.

(2) Branch lines are smaller diameter pipes distributing water from the mains to groups of consumers on secondary streets.

(3) Service piping is the final small diameter piping entering each building.

TABLE 4.4

TERTIARY PIPING SIZES

Load Center Heat Exchanger (see Fig. 4.2)	Nominal Pipe O.D., inches Main	Branch	Service
5	6	4	2
6	6	4	2
7	6	4	2
8	6	4	2
9	4	3	2
10	6	4	2
11	6	4	2
12	6	4	2
13	6	4	2
14	4	3	2
15	6	4	2
16	4	3	2
17	6	4	2
18	4	3	2
19	8	4	2
20	8	4	2
21	4	3	2
22	6	4	2

service water and are cooled by liquid absorption chillers. Heat for domestic hot water consumption is provided from additional heat exchangers. The coefficients of performance (efficiencies) of the domestic hot water and space heating systems are assumed to be unity, while the COP for the absorption units is nominally set at 75% and is varied with the internal room and ambient air temperatures according to theoretical Carnot cycle performance, scaled to reflect observed behavior at the set point temperature conditions. Table 4.5 shows the resulting design energy demands at each of the 22 heat exchangers in the utility system for a winter air temperature of 10 °F occurring at midnight and a summer temperature of 100 °F at noon. Also shown are the desired supply and demand side thermal utility system water temperatures and the design heat transfer coefficient for each of the heat exchangers. (In Appendix C are presented the detailed design criteria and component parameters used in sizing these heat exchangers.) The TDIST models assume the use of a single-pass, counterflow, straight tube heat exchanger configuration, and these constraints are used in sizing physically the units modelled in the computer simulations. However, the only limitations placed upon the actual heat exchangers chosen for the utility system are that they exhibit the required heat transfer characteristics and that their frictional

TABLE 4.5

100% THERMAL CASE HEAT EXCHANGER DESIGN PARAMETERS

Loop (see Fig. 4.2)	Heat Exchanger (see Fig. 4.2)	Winter Load (BTU/hrx10^6)	Summer Load (BTU/hrx10^6)	Supply Side $\Delta T(^\circ F)$	Demand Side $\Delta T(^\circ F)$	UA Value (BTU/hr°Fx10^4)
Primary	1	136.02	219.78	380-335	360-110	260.3
	2	156.14	230.77	335-285	315-110	327.2
	3	118.52	176.70	285-250	265-110	281.7
	4	191.71	305.55	250-185	240-110	938.9
1	5	51.34	71.24	360-110	240-100	160.9
	6	33.16	67.24	360-110	240-100	160.9
	7	24.00	38.29	360-100	240-100	97.2
	8	27.52	43.01	360-110	240-100	97.2
2	9	20.89	33.46	315-110	240-100	103.7
	10	40.19	62.49	315-110	240-100	214.9
	11	44.68	69.33	315-110	240-100	214.9
	12	50.38	65.49	315-110	240-100	214.9
3	13	29.48	56.52	265-110	220-100	242.9
	14	25.16	18.99	265-110	220-100	141.1
	15	21.63	32.84	265-110	220-100	141.1
	16	14.55	24.29	265-110	220-100	141.1
	17	27.70	44.06	265-110	220-100	242.9
4	18	19.10	29.24	240-110	220-100	202.7
	19	55.89	90.45	240-110	220-100	629.6
	20	56.24	90.83	240-110	220-100	629.6
	21	17.20	28.62	240-110	220-100	202.7
	22	43.28	66.41	240-110	220-100	629.6
Total System		602.39	932.80			

pressure losses at the design system fluid flowrates do not significantly alter the overall choice of the system and piping to be presented in Chapter 6.

4.2 85% Thermal Supply Option

In order to determine the optimum thermal-to-electrical energy demand ratio affording a minimum installed capacity for the power plant and a minimum total energy system cost over life, it is necessary to investigate a range of utility system options in addition to the limiting 100% case described above. To provide a consistent definition for these alternative systems, it has been decided to use the energy demands at the load center heat exchangers for the design winter day (see Table 4.5) as a common benchmark for all thermal load shedding. To simplify the analysis and to minimize the amount of system re-design required for every option studied, it was further decided to reduce the thermal utility system demands on an incremental load center heat exchanger basis and to minimize the total primary and secondary distribution loop piping for each option.* Thus,

*A finely detailed thermal load shedding scheme designed to optimize the utility system cost would not proceed according to this simplified incremental reduction formula, but would provide for each supply option that piping configuration which minimized the total capital cost of the thermal utility system per unit of thermal demand served. The systems discussed here approximate this type of analysis for the primary and secondary loop piping but do not fully consider the impacts of the tertiary piping network or non-incremental system reductions.

the nominal "85% Thermal" supply case consists of a thermal utility system whose peak winter design load is approximately 85% of the winter design thermal load for the 100% Thermal case. ("Approximately" because of the incremental nature of the load shedding; the actual measured loads are 85.2% of the 100% design demands).

4.2.1 Primary and Secondary Loop Piping

Figure 4.3 shows the primary and secondary piping system for the 85% Thermal case resulting in a minimum total primary and secondary piping distance. To facilitate cross-referencing between this system and the 100% Thermal case shown in Fig. 4.2, the heat exchanger indexing numbers are retained from the 100% system. The primary loop remains the same as in the preceding case except for the removal of heat exchanger #3, which had served secondary loop #3. Heat exchangers #14-17 and their associated secondary and tertiary piping have been removed, and their consumers' HTW-powered space conditioning and domestic hot water supply equipment has been replaced with heat pumps, compressive air conditioners and electric hot water heaters. Heat exchanger #13, whose load center is retained in the 85% thermal utility system, has been added to secondary loop #1. The building distribution within each load center remains the same as in Table 4.1. Except for the removal of loop #3 and the addition of 0.1 mile of pipe

Figure 4.3. "85% Thermal" Case Utility System Piping Layout

2

to loop #1, the nominal pipe and loop specifications for the 85% case are identical to those for the 100% case shown in Table 4.2. Similarly, but for the elimination of the piping associated with load center heat exchangers #14-#17, the 85% tertiary piping system is also identical to that summarized in Tables 4.3 and 4.4.

4.2.2 Heat Exchangers

As in the 100% case, all buildings served by the thermal utility system in the 85% case are equipped with hot water heat, absorption air conditioners and thermally-supplied domestic hot water. Those consumers not connected to the thermal distribution network (the buildings formerly served by heat exchangers #14-#17) have their space conditioning requirements supplied by heat pumps and compressive air conditioning units (heat pumps in the cooling mode of operation) and their domestic hot water needs provided from electric hot water heaters. The water heater COP's are assumed to be unity. The nominal COP of the heat pumps is set at 2.4, and that of the air conditioners is 2.0. As with the absorption units' COP's, these conversion efficiencies are varied according to the theoretical Carnot cycle temperature difference law modified to yield observed unit performance at the design set temperatures.* Table 4.6 summarizes the design summer and winter loads, supply and demand side temperatures and heat transfer coefficients for the 17 heat exchangers in the 85% case system. Appendix C contains the design criteria and component specifications used in sizing these heat exchangers.

4.3 75% Thermal Supply Option

Figure 4.4 shows the piping layout chosen for the nominal "75% Thermal" supply case utility system. (The actual measured design peak winter thermal demands for the load centers in the 75% system are 76.7% of the demands in the 100% case). The heat exchanger index numbers shown in the Figure correspond to those used in the 100% case (Fig. 4.2) to allow simple cross-reference of the two utility systems. The reduction in the thermal demands between the 85% and 75% systems is accomplished by the removal of heat exchangers #7 and #8 and the conversion of their thermal energy consumers to electrical space conditioning and domestic hot water service equipment (see Section 4.2.2). Table 4.7 summarizes the primary and secondary piping designs used in this utility system's simula-

*Automatic sensing circuits for the heat pumps are assumed to transfer operation to electric resistance heaters when the heat pumps' COP's fall below 1.0, but this condition is not reached on the given design day.

TABLE 4.6

85% THERMAL CASE HEAT EXCHANGER DESIGN PARAMETERS

Loop (see Fig. 4.3)	Heat Exchanger (see Fig. 4.3)	Winter Load (BTU/hrx10^6)	Summer Load (BTU/hrx10^6)	Supply Side ΔT(°F)	Demand Side ΔT(°F)	UA Value (BTU/hr°Fx10^4)
Primary	1	165.50	278.51	375-305	355-120	374.1
	2	156.14	232.09	305-250	285-120	388.2
	4	191.71	312.47	250-175	240-110	1085.0
1	5	51.34	72.12	355-120	240-100	132.8
	6	33.15	68.57	355-120	240-100	132.8
	7	24.00	28.29	355-125	240-100	79.2
	8	27.52	43.01	355-120	240-100	79.2
	13	29.48	56.52	355-120	240-100	132.8
2	9	20.89	33.46	285-120	240-100	108.5
	10	40.19	62.49	285-120	240-100	224.9
	11	44.68	69.33	285-120	240-100	224.9
	12	50.38	66.81	285-120	240-100	224.9
4	18	19.10	29.24	240-110	220-100	202.7
	19	55.89	94.43	240-110	220-100	654.5
	20	56.24	93.77	240-110	220-100	654.5
	21	17.20	28.62	240-110	220-100	202.7
	22	43.28	66.41	240-110	220-100	654.5
Total System		513.35	823.07			

TABLE 4.7

FORT BRAGG 75% THERMAL CASE PIPE PARAMETERS

Loop[1]	Total Length	Pipe O.D.[2]	Insulation[3]
1	2.4 miles	12"	Type 1[4]
2	7.8 miles	12"	Type 1[5]
4	6.2 miles	12"	Type 1[6]
Primary	10.0 miles	18"	Type 2[7]

(1) See Fig. 4.4.

(2) Nominal initial simulation diameters. Final values of optimum system set according to frictional losses as discussed in Chapter 6.

(3) Type 1 insulation (12" pipe): 3" calcium silicate insulation, 1" air space, 10 gauge galvanized steel spiral welded conduit, asphalt impregnated fiberglass screen, fiberglass reinforced asbestos pipe line felt, buried 6 ft. deep on center. (Meets Army Corps of Engineers specification CE-301.21)

Type 2 insulation: (18" pipe): 4" calcium silicate insulation, 1" air space, 10 gauge galvanized steel spiral welded conduit, asphalt impregnated fiberglass screen, fiberglass reinforced asbestos pipe line felt, buried 6 ft. deep on center. (Meets Army Corps of Engineers specification CE-301.21)

(4) Return piping uninsulated, approximately 1.2 miles

(5) Return piping uninsulated, approximately 3.9 miles

(6) Return piping uninsulated, approximately 3.1 miles

(7) Return piping from secondary heat exchanger #4 to power plant uninsulated, approximately 3.1 miles

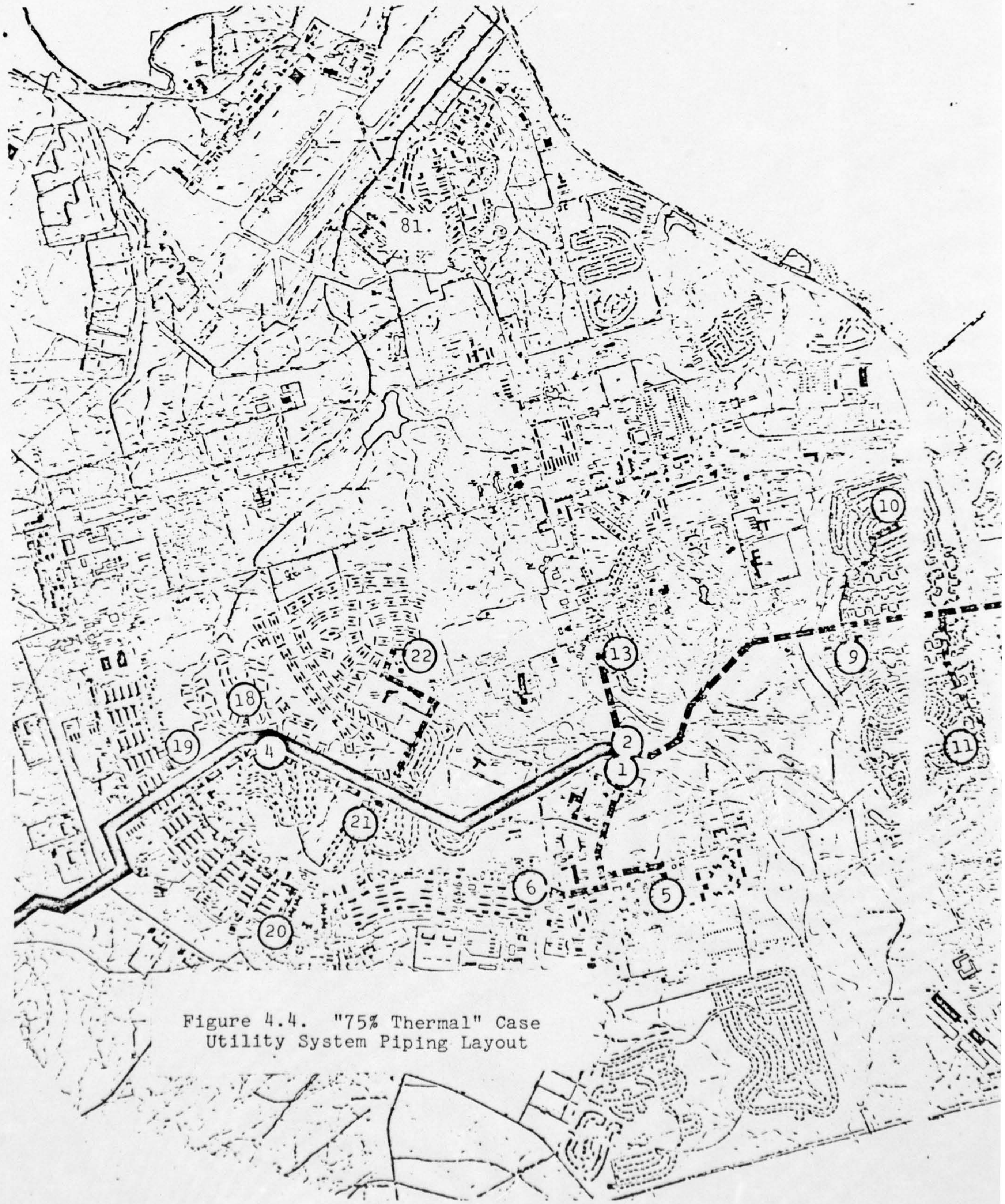


Figure 4.4. "75% Thermal" Case
Utility System Piping Layout


N
10
13
9
12
2
1
11
6
5
FORT BRAGG
MASTER PLAN
BUILDING AREA

tions. The tertiary piping remains the same as that shown in Tables 4.3 and 4.4 for the retained load center heat exchangers. Table 4.8 shows the design loads, desired operating temperatures and heat transfer coefficients for the 15 heat exchangers in the 75% thermal utility system, and Appendix C contains the detailed design parameters used in sizing each of the units.

4.4 65% Thermal Supply Option

Eliminating load center heat exchangers #6 and #13 and moving heat exchanger #5 to secondary loop #2 allows the reduction of the 75% case thermal utility system demands to 66.3% of the 100% case winter peak design loads. With the removal of secondary loop #1 and its heat exchanger, the "65% Thermal" case utility system is designed as shown in Fig. 4.5. As in the preceding options, the buildings removed from the thermal utility system (see Table 4.1) are provided with electrical end-use equipment to meet their space conditioning and hot water demands. Except for the elimination of secondary loop #1 and the addition of 1.9 miles of pipe to loop #2 to supply heat exchanger #5, the primary and secondary loop piping designs are the same as those for the 75% case summarized in Table 4.7. Tertiary piping, except for those load centers removed from the thermal distribution network, remains identical to that

TABLE 4.8

75% THERMAL CASE HEAT EXCHANGER DESIGN PARAMETERS

Loop (see Fig. 4.4)	Heat Exchanger (see Fig. 4.3)	Winter Load ($BTU/hr \times 10^6$)	Summer Load ($BTU/hr \times 10^6$)	Supply Side $\Delta T(^\circ F)$	Demand Side $\Delta T(^\circ F)$	UA Value ($BTU/hr^\circ F \times 10^4$)
Primary	1	113.98	195.00	380-320	360-120	249.4
	2	156.14	230.77	320-250	300-120	392.7
	4	191.71	305.55	250-155	240-110	1295.0
1	5	51.34	71.24	360-120	240-100	127.6
	6	33.16	67.24	360-120	240-100	127.6
	13	29.48	56.52	360-120	240-100	127.6
2	9	20.89	b3.46	300-120	240-100	91.9
	10	40.19	62.49	300-120	240-100	190.4
	11	44.68	69.33	300-120	240-100	190.4
	12	50.38	65.49	300-120	240-100	190.4
4	18	19.10	29.24	240-110	220-100	202.7
	19	55.89	90.45	240-110	220-100	629.6
	20	56.24	90.83	240-110	220-100	629.6
	21	17.20	28.62	240-110	220-100	202.7
	22	43.28	66.41	240-110	220=100	692.6
Total System		461.83	731.32			

Figure 4.5. "65% Thermal" Case
Utility System Piping Layout

listed in Tables 4.3 and 4.4. The major heat exchanger design parameters for the 65% system are shown in Table 4.9, and additional heat exchanger component details are presented in Appendix C.

TABLE 4.9

65% THERMAL CASE HEAT EXCHANGER DESIGN PARAMETERS

Loop (see Fig. 4.5)	Heat Exchanger (see Fig. 4.5)	Winter Load (BTU/hrx10^6)	Summer Load (BTU/hrx10^6)	Supply Side ΔT(°F)	Demand Side ΔT(°F)	UA Value (BTU/hr°Fx10^4)
Primary	2	207.48	304.21	375-270	355.120	471.5
	4	191.71	312.47	270-160	250-110	933.0
2	5	51.34	72.12	355-120	240-100	132.8
	9	20.89	33.46	355-120	240-100	61.6
	10	40.19	62.49	355-120	240-100	132.8
	11	44.68	69.33	355-120	240-100	132.8
	12	50.38	66.81	355-120	240-100	132.8
4	18	19.10	29.24	250-110	220-100	160.6
	19	55.89	94.43	250-110	220-100	518.7
	20	56.24	93.77	250-110	220-100	518.7
	21	17.20	28.62	250-110	220-100	160.6
	22	43.28	66.41	250-110	220-100	518.7
Total System		399.19	616.68			

CHAPTER 5

FORT BRAGG UTILITY SYSTEM SIMULATION RESULTS

In the preceding two chapters, the discussions of the choice of the eleven Fort Bragg energy consumer categories and of the design criteria and layouts of the four thermal utility system options have been limited to instantaneous energy demand conditions calculated for the base during the summer and winter peak design days. The basic data required to determine the optimal utility system design-affording the minimum total cost over its lifetime -- are the installed power plant thermal and electrical power generation capacities, the thermal energy storage reservoir capacity, the energy loss and sizing criteria for the thermal distribution network piping and the total energy produced annually by the power plant. Therefore, in order to determine these economic study input data, and to investigate the behavior and stability of the thermal utility system over a range of seasonally-varying thermal and electrical energy demand schedules, several computer simulations, each covering a 24-hour period, have been performed. These calculations involved varying the energy supply system options for the base during six different days throughout the year. After studying weather data for Fort Bragg obtained from recent NOAA Fayettville weather station records and historically averaged temperature data for the adjacent Simmons Air Force Base (provided by the Department of the Air Force [1]), it has been determined that the seasonal weather conditions at this location are approximately symmetrical between the spring and autumn. The days chosen for study have thus been designated as the peak winter heating demand day, an average winter day, an early spring day (identical to a late fall day), a late spring (early fall) day, an average summer day, and the peak summer cooling demand day. Because of their importance in determining the system design parameters, the simulations for the summer and winter peak days have been performed for all four utility system configurations described in Chapter 4. As is discussed subsequently, it was discovered early in this analysis that the cases of interest in determining the optimal system design lie within the range of 65% to 85% in thermal/electrical load split values and the remainder of the daily simulations are limited to the three system options lying within this range. Due to time and resource limitations, no detailed design or annual demand analyses have been performed for thermal/electrical load split values below the nominal 65% case.

As is described in Chapter 4, the "non-space-conditioning" electrical demands for Fort Bragg during a 24-hour period have been obtained from central metering at the base [2]. To eliminate the contributions of electrical heating and cooling demands in this load schedule, the minimum weekday demand schedule (occurring on March 26, 1975) has been assumed to reflect little or no influence from space-conditioning equipment, and this schedule is used to specify the constant non-space-conditioning electrical load component of the base's energy demands in each of the daily simulations. Table 5.1 lists this demand schedule.

5.1 Daily Thermal and Electrical Energy Demand Schedules

5.1.1 Winter Peak Heating Demand Day

The weather conditions for the peak winter heating day have been described in Chapter 3 and will be summarized briefly here for completeness. Figure 5.1 illustrates the air temperature profile for this day. Although NOAA records indicate that the lowest temperature recorded at Fayetteville during the past 40 years is 5 °F, the 15 °F minimum chosen for the simulation day has been found to be typical of expected annual winter extremes. The minimum and maximum temperatures shown were obtained from NOAA daily records for 1975, and the peak-day temperature profile has been fitted to these extremes using hourly temperature data from Boston's Logan Airport weather station. Winds are assumed to remain constant at 15 mph from the west throughout the day. A peak solar radiation intensity of 390 BTU/hr per horizontal square foot of surface area is assumed to obtain under cloudless skies, and this value is modified by seasonal atmospheric absorption and diffusion effects (see Table 5.2). Appendix B describes the domestic hot water and electrical

TABLE 5.1

FORT BRAGG NON-SPACE-CONDITIONING ELECTRICAL POWER DEMANDS

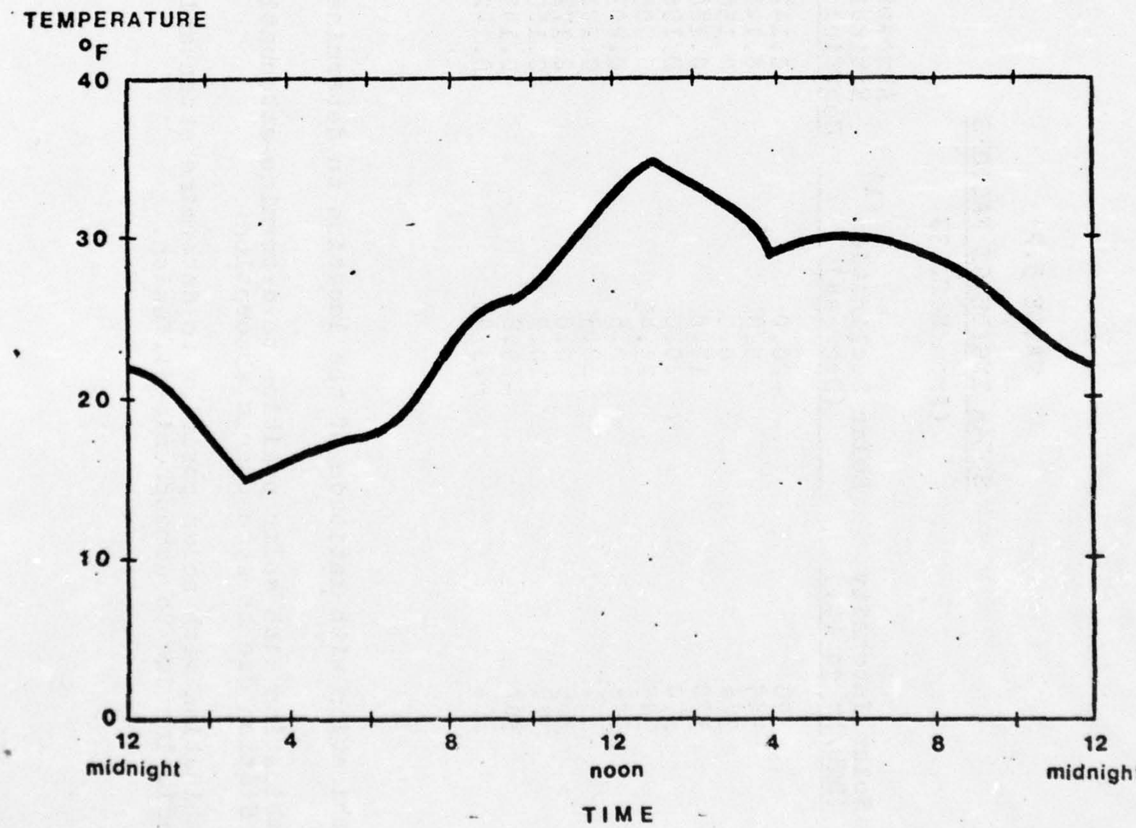
(From Ref. 2)

Time[1]		Demand (MWe)
12		19.60
1		18.90
2		17.85
3		17.50
4		17.85
5		18.20
6	AM	19.25
7		22.05
8		27.30
9		30.45
10		31.50
11		31.50
12		30.80
1		30.45
2		29.75
3		29.40
4		28.70
5		28.70
6	PM	29.40
7		31.50
8		35.00
9		33.95
10		30.10
11		26.25
12		19.60

[1] Times shown are Eastern Standard Time; the demand schedule has been shifted by one hour during summer months to account for Daylight Savings Time.

Figure 5.1

Winter Peak Heating Demand Day Air Temperatures



AD-A031 744 MASSACHUSETTS INST OF TECH CAMBRIDGE DEPT OF NUCLEAR--ETC F/G 13/1
DESIGN OF A NUCLEAR POWERED TOTAL ENERGY SYSTEM FOR FORT BRAGG,--ETC(U)
MAY 76 J W STETKAR, F R BEST, M W GOLAY DAAK02-74-C-0308

UNCLASSIFIED USAFESA-RT-2011 NL

2 OF 4
AD
A031744

TABLE 5.2

SOLAR INCIDENCE FACTORS

(from Ref. 3)

Date	Solar Intensity (BTU/hr.sq.ft.)	Solar Declination[(1)] (Degrees)	Atmospheric Extinction Coefficient[(2)]	Sky Diffusion Factor[(3)]
Jan. 21	390	-20.0	0.142	0.058
Feb. 21	385	-10.8	0.144	0.060
Mar. 21	376	0.0	0.156	0.071
Apr. 21	360	11.6	0.180	0.097
May 21	350	20.0	0.196	0.121
June 21	345	23.45	0.205	0.134
July 21	344	20.6	0.207	0.136
Aug. 21	351	12.3	0.201	0.122
Sept. 21	365	0.0	0.177	0.092
Oct. 21	378	-10.5	0.160	0.073
Nov. 21	387	-19.8	0.149	0.063
Dec. 21	391	-23.45	0.142	0.057

(1) Used in combination with latitude of the location to determine solar position.

(2) Used in combination with solar position to determine attenuation of normal incident radiation due to atmospheric absorption.

(3) Used in combination with solar position to determine attenuation of normal incident radiation due to atmospheric diffusion.

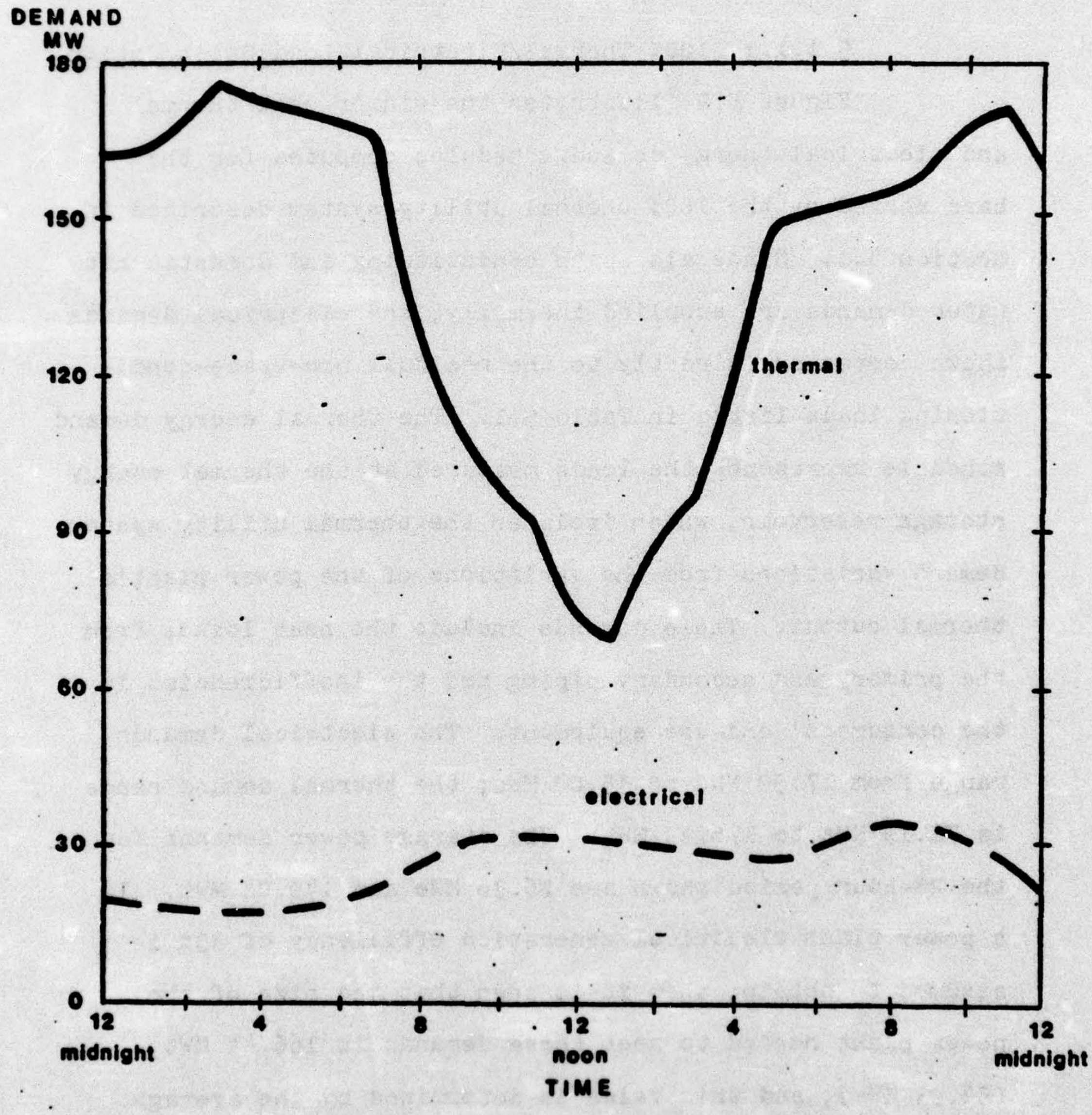
equipment use factors and the desired room temperatures for each of the eleven consumer categories for this day.

5.1.1.1 <u>100% Thermal/Electrical Load Split Option</u>

Figure 5.2 illustrates the winter peak thermal and electrical energy demand schedules computed for the base served by the 100% thermal utility system described in Section 4.1. Since all space conditioning and domestic hot water demands are supplied thermally, the electrical demands shown correspond directly to the residual non-space-conditioning loads listed in Table 5.1. The thermal energy demand schedule represents the loads measured at the thermal energy storage reservoir, which isolates the thermal utility system demand variations from the variations of the power plant's thermal output. These demands include the heat losses from the primary and secondary piping and the inefficiencies in the consumers' end-use equipment. The electrical demands range from 17.50 MWe to 35.00 MWe; the thermal demand range is 71.19 MWt to 176.43 MWt. The average power demands for the 24-hour period shown are 26.36 MWe and 140.08 MWt. If a power plant electrical generation efficiency of 33% is assumed to obtain, then it is seen that the size of the power plant needed to meet these demands is 166.44 MWt (54.93 MWe), and this value is determined by the average thermal load (i.e., meeting the peak electrical demand at

Figure 5.2

Peak Winter Day
100% Thermal/Electrical Load Split
Energy Demand Schedules



a generation efficiency of 33% would require a power plant thermal rating of 106.06 MWt, but this plant would be undersized for the 140.08 MWt average thermal demand. Operating at rated capacity, the plant would produce an average thermal output of 106.06 MWt - 26.36 MWe = 79.70 MWt. On the other hand, a power plant with a thermal capacity of 166.44 MWt, combined with the load smoothing capability of the interfacing reservoir, would just meet the base's thermal loads and, with its generation efficiency depending upon the nature of the plant control scheme chosen, would also supply the given electrical demands). If the 166.44 MWt power plant were used to supply the given demand schedules by operating constantly at its peak total energy output rate, the HTW thermal energy storage reservoir (sized to smooth the supply and demand imbalances) would require a total volume of 97,604 cubic feet (730,026 gallons) and could be contained within a right circular cylinder having a diameter and height of 49.90 feet.*

The main features of the TUS load behavior during cold weather are apparent in this simulation. It is seen that the thermal load maximum is encountered during the early morning hours - when the air is coldest, and when

*Because of the high pressures required to maintain the 380°-400 °F HTW in its liquid form (approximately 250 psi), the actual reservoir would most probably consist of a set of several smaller cylindrical pressurized tanks. The single cylinder dimensions given here and in the following sections are presented only as general references for comparison with other large-volume containers.

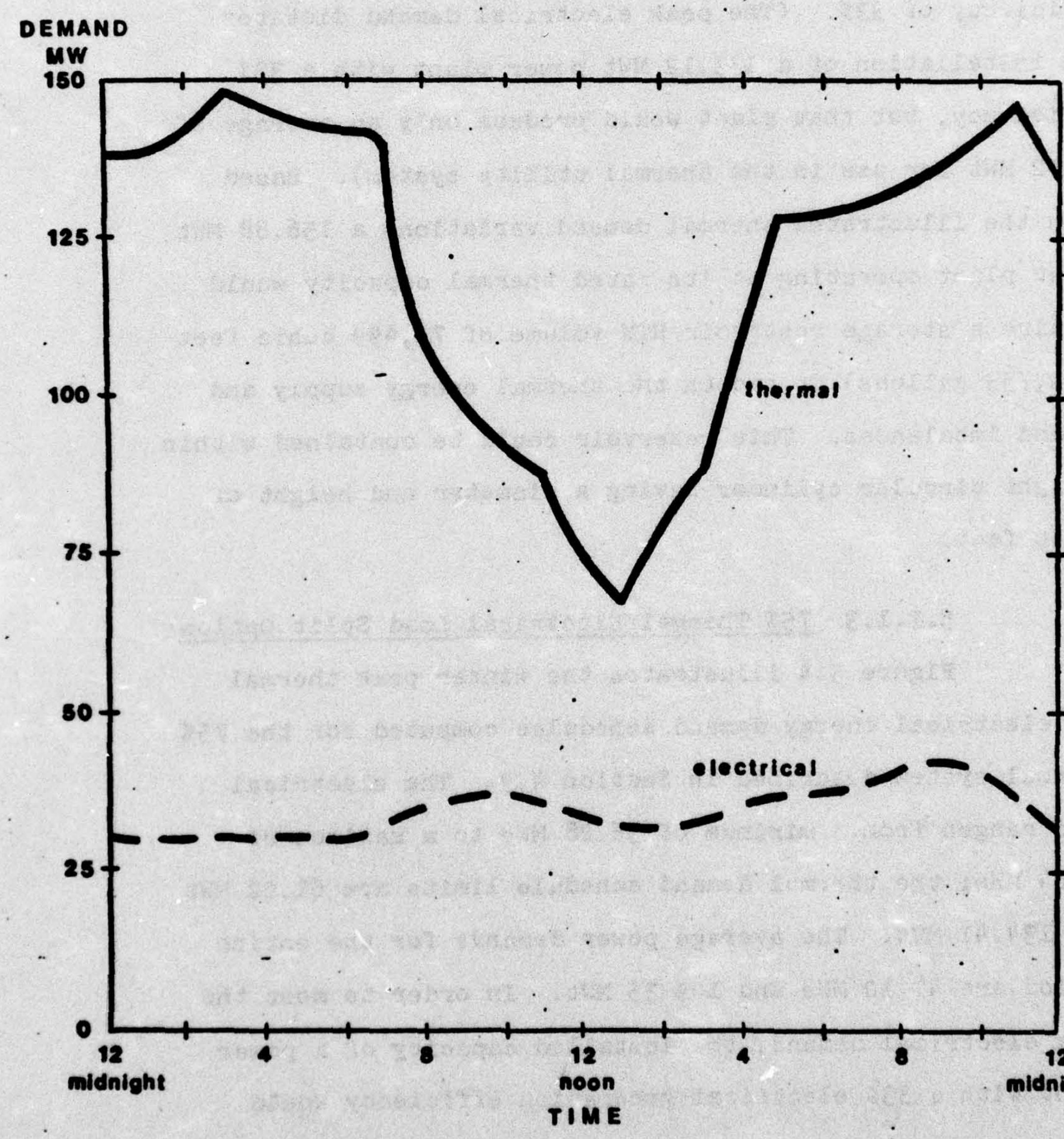
the sun is not shining. During the subsequent hours - as the effect of solar heating becomes more important - the thermal power TUS load declines to a minimum in the early afternoon, from which it then increases steadily during the late afternoon and early evening hours. It is seen also in cases in which the electrical space-coniditoning load is significant that the electrical demand schedule follows the same sort of daily schedule.

5.1.1.2 85% Thermal/Electrical Load Split Option

Figure 5.3 shows the thermal and electrical energy demand schedules computed for the 85% thermal/electrical load split system during the peak winter demand day. With the addition of electric hot water heaters and heat pumps to the buildings not served by the thermal energy piping network, the electrical demands are seen to have increased (from the 100% split case) to a minimum of 30.69 MWe and a maximum of 43.93 MWe. The shape of the demand profile is also changed somewhat from that of the 100% simulation shown in Fig. 5.2 due to the influence of the large space conditioning loads in the evening hours. The thermal utility system demands have been reduced to a minimum of 69.17 MWt and a maximum of 149.65 MWt. The average power demands for the 24-hour period are 35.40 MWe and 121.48 MWt, and the power plant capacity required to supply these demands is 156.88 MWt

Figure 5.3

Peak Winter Day
85% Thermal/Electrical Load Split
Energy Demand Schedules



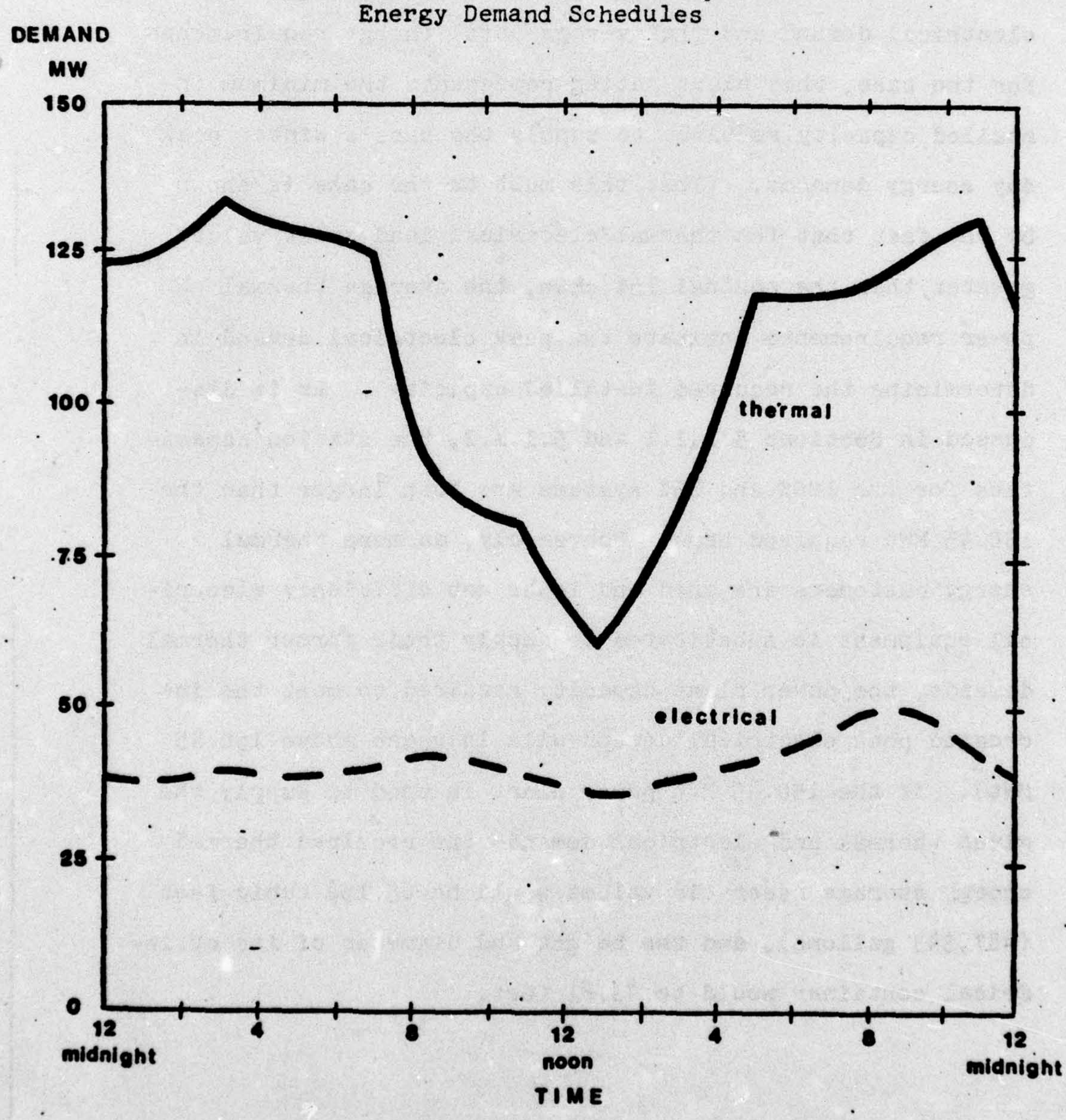
(51.77 MWe). As in the 100% case, this capacity is determined by the requirement of meeting the average thermal demand with a plant having a maximum electrical generation efficiency of 33%. (The peak electrical demand dictates the installation of a 133.12 MWt power plant with a 33% efficiency, but that plant would produce only an average of 97.72 MWt for use in the thermal utility system). Based upon the illustrated thermal demand variations a 156.88 MWt power plant operating at its rated thermal capacity would require a storage reservoir HTW volume of 73,499 cubic feet (549,733 gallons) to smooth the thermal energy supply and demand imbalances. This reservoir could be contained within a right circular cylinder having a diameter and height of 45.40 feet.

5.1.1.3 <u>75% Thermal/Electrical Load Split Option</u>

Figure 5.4 illustrates the winter peak thermal and electrical energy demand schedules computed for the 75% thermal system described in Section 4.3. The electrical load ranges from a minimum of 36.28 MWe to a maximum of 49.64 MWe; the thermal demand schedule limits are 61.02 MWt and 134.41 MWt. The average power demands for the entire period are 41.10 MWe and 109.35 MWt. In order to meet the peak electrical demand, the installed capacity of a power plant with a 33% electrical generation efficiency would

Figure 5.4

Peak Winter Day
75% Thermal/Electrical Load Split
Energy Demand Schedules

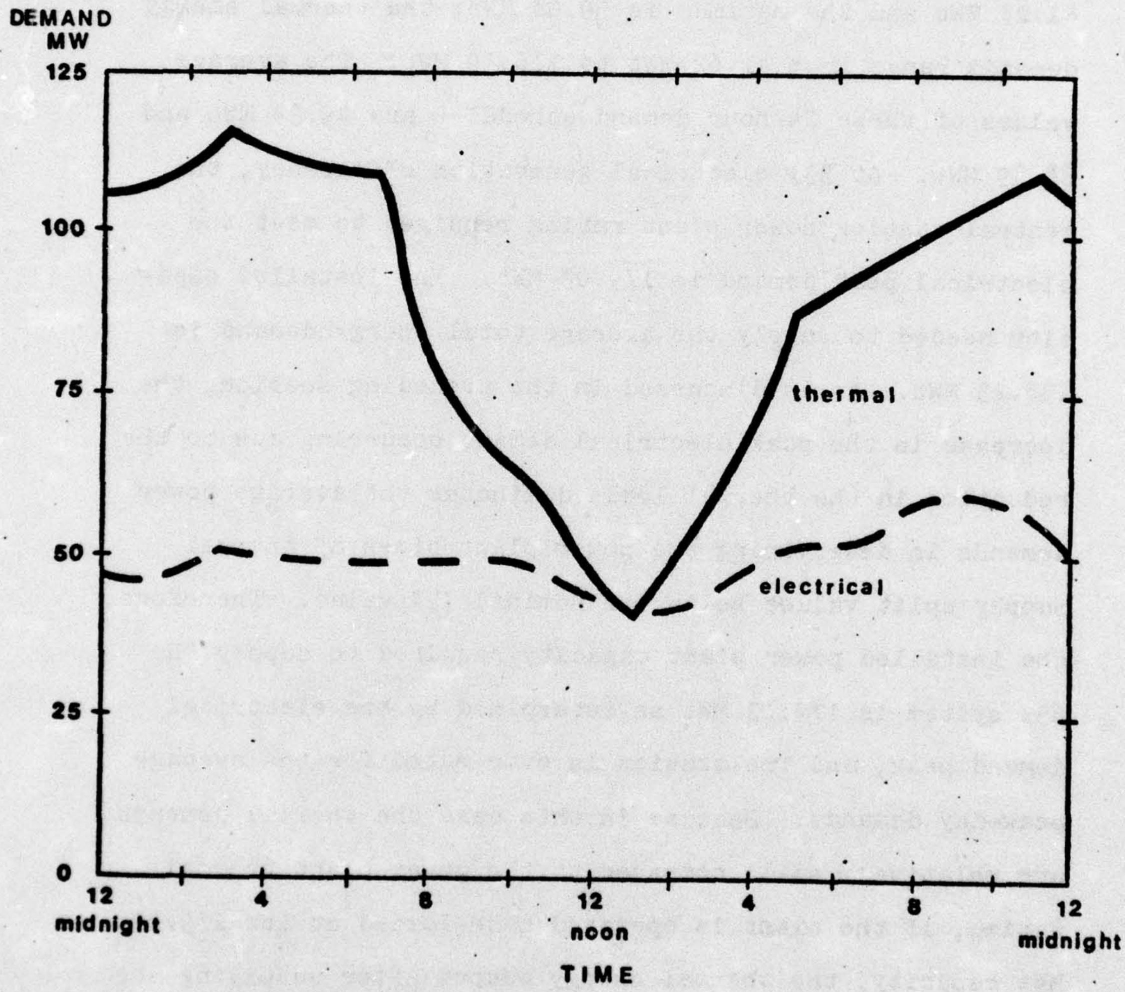


have to be 150.42 MWt. The capacity required to meet the average total energy demands for the base is 150.45 MWt. Since the 150.45 MWt power plant would meet both the peak electrical demand and the average total energy requirements for the base, that plant rating represents the minimum installed capacity required to supply the base's winter peak day energy demands. (That this must be the case is shown by the fact that for thermal/electrical load split values greater than the nominal 75% case, the average thermal power requirements dominate the peak electrical demand in determining the required installed capacity . As is discussed in Sections 5.1.1.1 and 5.1.1.2, the station capacities for the 100% and 85% systems are both larger than the 150.45 MWt required here. Conversely, as more thermal energy customers are shed and lower net efficiency electrical equipment is substituted to supply their former thermal demands, the power plant capacity required to meet the increased peak electrical demand will increase above 150.45 MWt). If the 150.45 MWt power plant is used to supply the given thermal and electrical demands the required thermal energy storage reservoir volume would be 65,158 cubic feet (487,343 gallons), and the height and diameter of its cylindrical container would be 43.61 feet.

5.1.1.4 65% Thermal/Electrical Load Split Option

The 65% thermal/electrical load split system winter peak thermal and electrical energy demand schedules are shown in Fig. 5.5. The minimum electrical demand is 41.27 MWe and the maximum is 59.08 MWe; the thermal energy demands range from 41.62 MWt to 116.19 MWt. The average values of these 24-hour demand schedules are 49.34 MWe and 88.89 MWt. At 33% electrical generation efficiency, the central station power plant rating required to meet the electrical peak demand is 179.03 MWt. The installed capacity needed to supply the average total energy demand is 138.23 MWt. As is discussed in the preceding section, the increase in the peak electrical demand occurring due to the reduction in the thermal loads dominates the average power demands in determining the power plant sizes at thermal supply split values below the nominal 75% value. Therefore, the installed power plant capacity required to supply the 65% system is 179.03 MWt as determined by the electrical demand peak, and the station is over-rated for the average peak-day demands. Because in this case the thermal demands are relatively small compared to the power plant thermal rating, if the plant is operated base-loaded at its 179.03 MWt capacity, the thermal energy output after supplying the electrical load is always greater than the thermal demands

Figure 5.5
Peak Winter Day
65% Thermal/Electrical Load Split
Energy Demand Schedules



shown for the 24-hour period. Therefore, insofar as the primary function of the HTW reservoir is to supply the thermal loads when the power plant output is less than the system demand, there is no need for a thermal storage reservoir if the power plant operates as assumed.*

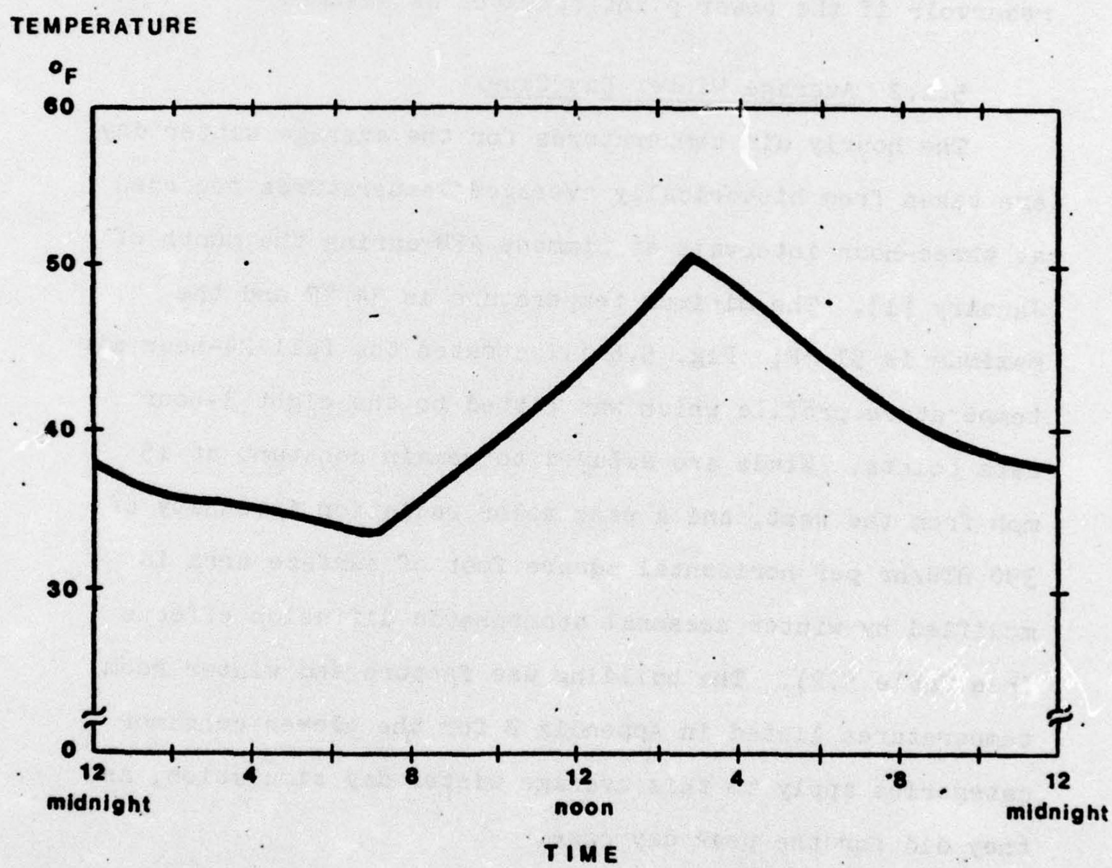
5.1.2 Average Winter Day Case

The hourly air temperatures for the average winter day are taken from historically averaged temperatures recorded at three-hour intervals at Simmons AFB during the month of January [1]. The minimum temperature is 34 °F and the maximum is 51 °F; Fig. 5.6 illustrates the full 24-hour air temperature profile which was fitted to the eight 3-hour data points. Winds are assumed to remain constant at 15 mph from the west, and a peak solar radiation intensity of 390 BTU/hr per horizontal square foot of surface area is modified by winter seasonal atmospheric diffusion effects (see Table 5.2). The building use factors and winter room temperatures listed in Appendix B for the eleven consumer categories apply to this average winter day simulation, as they did for the peak day case.

*Of course, under these conditions, some form of auxiliary cooling system would be required in order to dissipate the 1000 MWhr of excess thermal energy produced during the 24 hour period.

Figure 5.6

Average Winter Day Air Temperatures



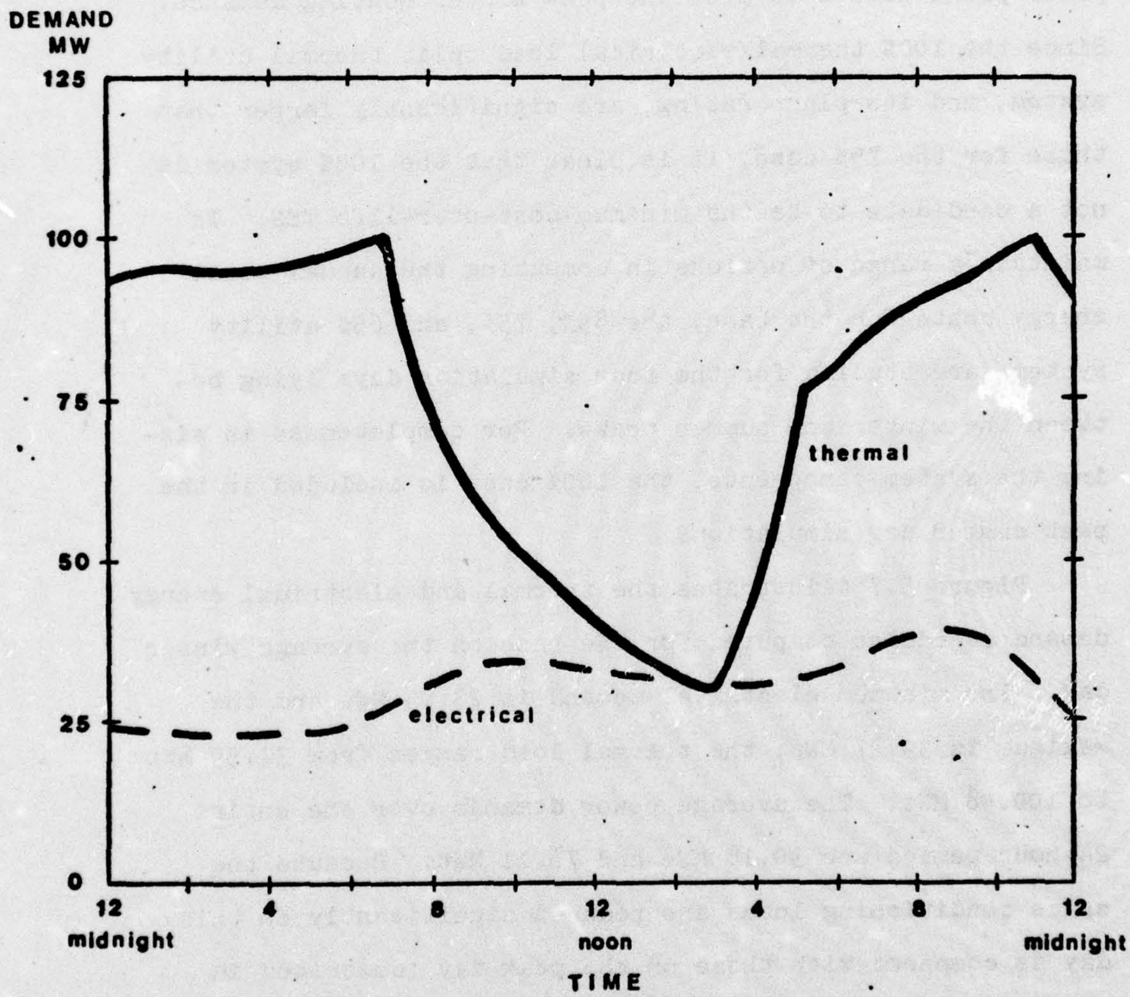
5.1.2.1 85% Thermal/Electrical Load Split Option

As is discussed in the preceding sections, the 75% thermal utility system affords the possibility for minimizing the installed capacity of the central station power plant needed to meet the peak winter heating demands. Since the 100% thermal/electrical load split thermal utility system, and its plant rating, are significantly larger than those for the 75% case, it is clear that the 100% system is not a candidate to be the minimum cost-over-life TES. To maintain a range of options in computing the annual total energy costs for the base, the 85%, 75%, and 65% utility systems are studied for the four simulation days lying between the winter and summer peaks. For completeness in sizing the system components, the 100% case is included in the peak summer day simulations.

Figure 5.7 illustrates the thermal and electrical energy demand schedules computed for the base on the average winter day. The minimum electrical demand is 23.05 MWe and the maximum is 39.27 MWe; the thermal load ranges from 31.59 MWt to 100.98 MWt. The average power demands over the entire 24-hour period are 30.18 MWe and 76.11 MWt. Because the space conditioning loads are reduced significantly on this day as compared with those on the peak day (described in Section 5.1.1.2), the electrical load peak is almost totally composed of the non-space-conditioning energy demands. The

Figure 5.7

Average Winter Day
85% Thermal/Electrical Load Split
Energy Demand Schedules



central power station capacity required to meet this peak at a 33% electrical generation efficiency is 119.00 MWt. The average total energy demands require a power plant rating of 106.29 MWt. From these two power plant ratings, it is seen that - due to the large annually-constant residual non-space-conditioning electrical load - on the average winter day for the 85% utility system the average thermal demands no longer dominate the plant sizing criteria as they did on the peak winter day. The plant must operate at 119.00 MWt to meet the electrical peak load, and it is over-rated for the average demands by approximately 13 MWt. This average discrepancy is not great enough to eliminate completely the need for a thermal energy storage reservoir, since at several times during the day the plant's constant 119.00 MWt energy output is too small to meet both the instantaneous electrical power and thermal utility system power demands. Therefore, the required HTW reservoir volume is 16,966 cubic feet (126,892 gallons), and it could be contained within a cylindrical tank with a diameter and height of 27.85 feet. The 311 MWhr of excess thermal energy generated in addition to that needed to supply the utility system and to re-charge the reservoir must be dissipated by an auxiliary cooling system.

5.1.2.2 75% Thermal/Electrical Load Split Option

The thermal and electrical energy demand schedules for the average winter day 75% thermal/electrical load split utility system are shown in Fig. 5.8. The electrical demand limits are 26.39 MWe and 42.12 MWe, and the thermal load ranges from 28.58 MWt to 91.06 MWt. The average power demands are 32.76 MWe and 68.52 MWt. As in the 85% thermal/electrical load split case, the 127.64 MWt plant capacity needed to meet the peak electrical demand at 33% generation efficiency dominates the 101.28 MWt average total energy demand in determining the magnitude of the central station's thermal power rating. However, in this case, the plant is so greatly over-rated that when operating at its rated total power output, its net thermal energy production always exceeds the instantaneous thermal demand, and no thermal storage reservoir is required. An auxiliary cooling system must also be used to dissipate the 646 MWhr of excess thermal energy produced over the 24-hour period.

5.1.2.3 65% Thermal/Electrical Load Split Option

Figure 5.9 illustrates the 65% utility system average winter day thermal and electrical energy demand schedules. The minimum electrical demand is 30.41 MWe; the maximum demand is 46.61 MWe. The thermal load schedule limits are 16.58 MWt and 77.73 MWt, and the 24-hour average power demands are 36.45 MWe and 54.59 MWt. The power plant rating required

Figure 5.8

Average Winter Day
75% Thermal/Electrical Load Split
Energy Demand Schedules

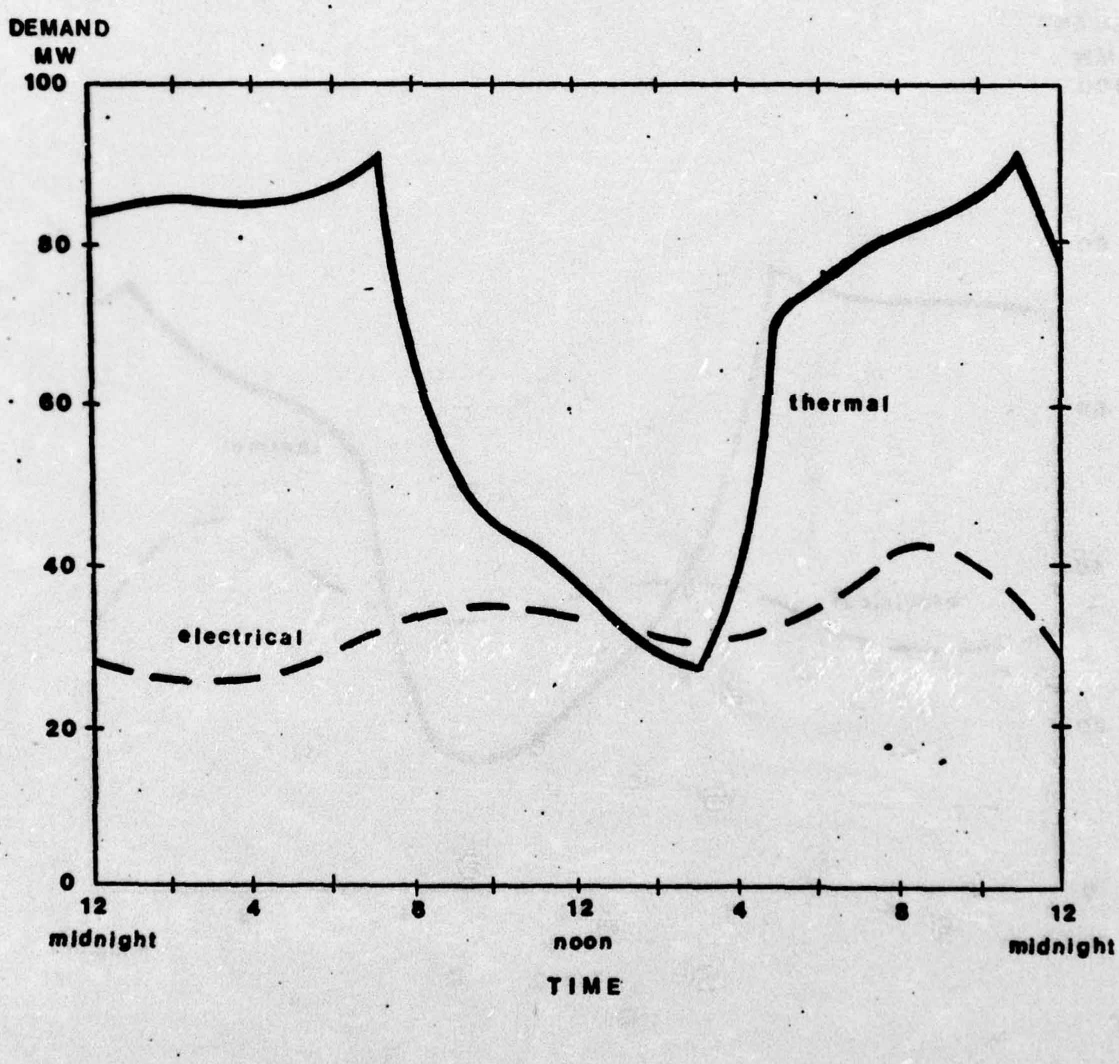
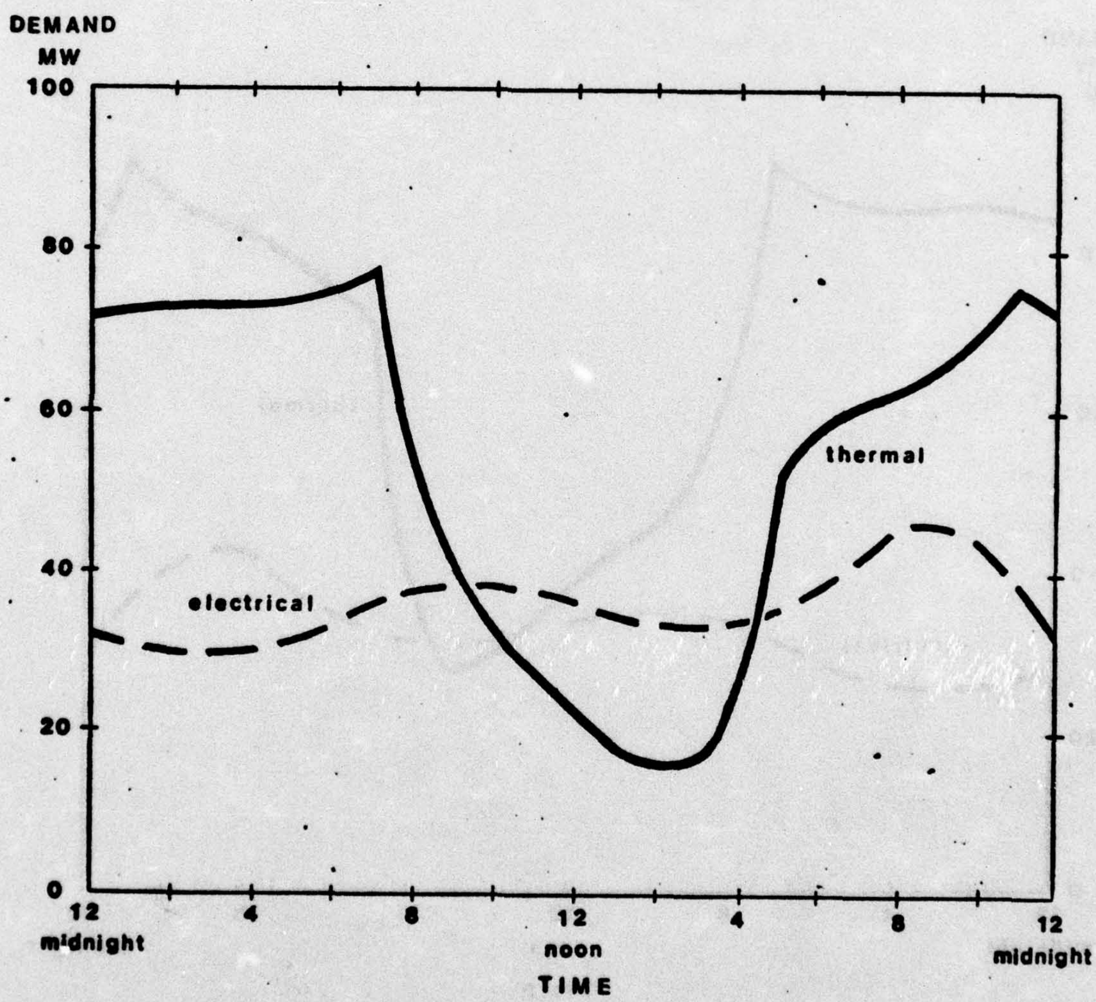


Figure 5.9

Average Winter Day
65% Thermal/Electrical Load Split
Energy Demand Schedules

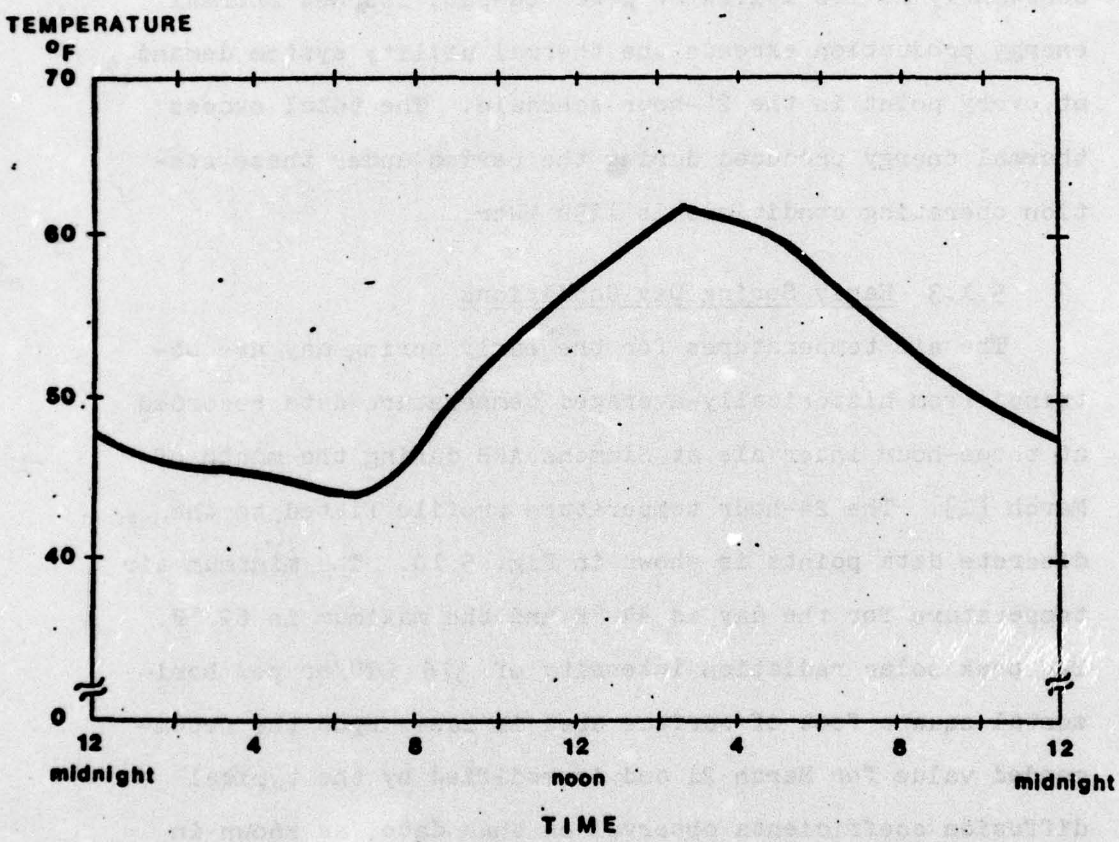


to meet the electrical peak demand at 33% generation efficiency is 141.24 MWt. (The average total energy demands require a plant capacity of 91.03 MWt.) Because of the relatively large electrical peak, if the power plant operates constantly at its 141.24 MW power output, its net thermal energy production exceeds the thermal utility system demand at every point in the 24-hour schedule. The total excess thermal energy produced during the period under these station operating conditions is 1230 MWhr.

5.1.3 Early Spring Day Conditions

The air temperatures for the early spring day are obtained from historically-averaged temperature data recorded at three-hour intervals at Simmons AFB during the month of March [1]. The 24-hour temperature profile fitted to the discrete data points is shown in Fig. 5.10. The minimum air temperature for the day is 44 °F and the maximum is 62 °F. The peak solar radiation intensity of 376 BTU/hr per horizontal square foot of surface area is based upon the recommended value for March 21 and is modified by the typical diffusion coefficients observed on that date, as shown in Table 5.2. Winds are assumed to remain constant at 15 mph from the west. The winter day room temperature settings and the building use factors presented in Appendix B apply directly to this day. Since the air temperatures for this

Figure 5.10
Early Spring Day* Air Temperatures
(*Identical to a Late Fall Day)

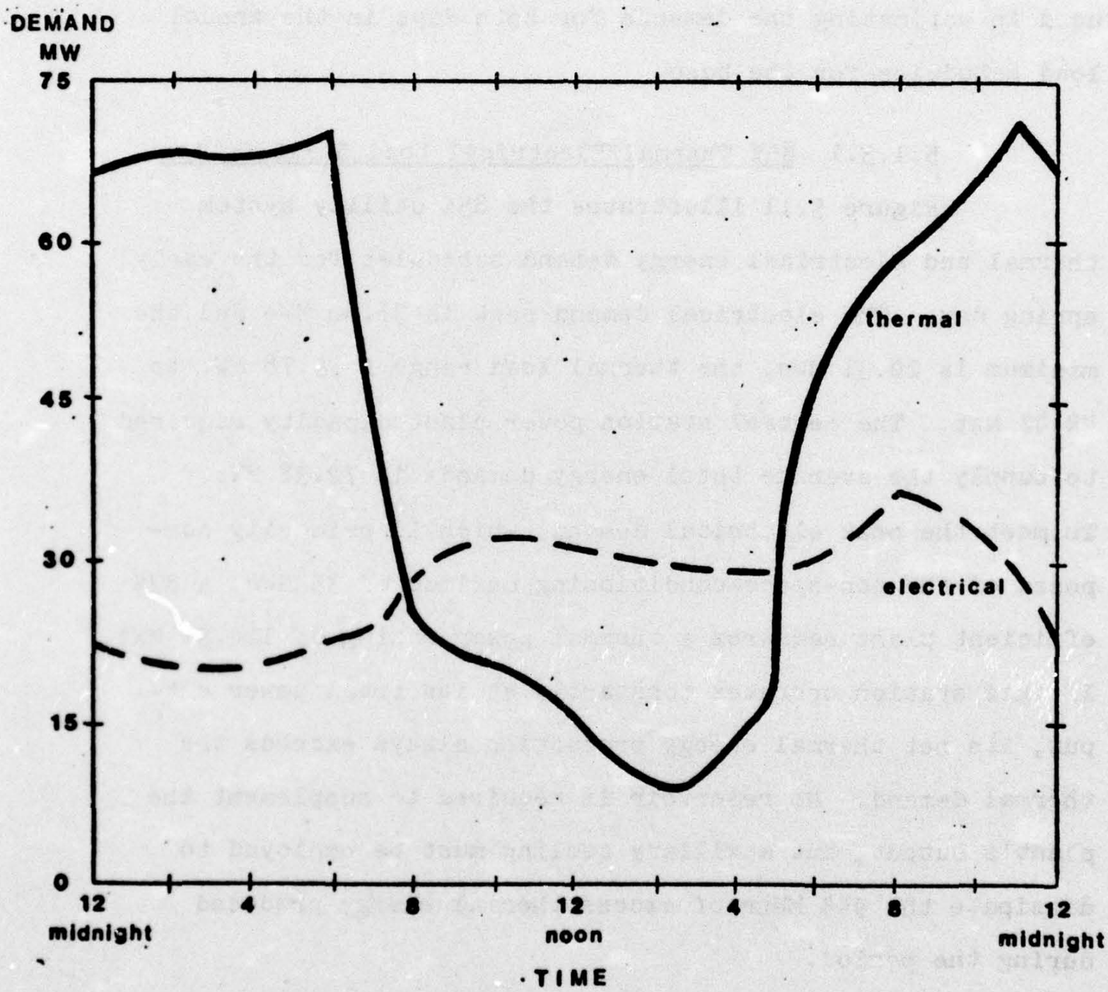


average early spring day correspond very closely to those recorded for an average November day, it is assumed that the base's space conditioning loads will be nearly identical on these two days, and the spring day simulations are used in estimating the demands for both days in the annual load schedules for the base.

5.1.3.1 85% Thermal/Electrical Load Split Option

Figure 5.11 illustrates the 85% utility system thermal and electrical energy demand schedules for the early spring day. The electrical demand peak is 36.60 MWe and the minimum is 20.31 MWe; the thermal load range is 9.78 MWt to 44.47 MWt. The central station power plant capacity required to supply the average total energy demands is 72.38 MWt. To meet the peak electrical demand, which is primarily composed of the non-space-conditioning maximum of 35 MWe, a 33% efficient plant requires a thermal power rating of 110.91 MWt. If this station operates constantly at its total power output, its net thermal energy production always exceeds the thermal demand. No reservoir is required to supplement the plant's output, but auxiliary cooling must be employed to dissipate the 944 MWhr of excess thermal energy produced during the period.

Figure 5.11
Early Spring Day
85% Thermal/Electrical Load Split
Energy Demand Schedules

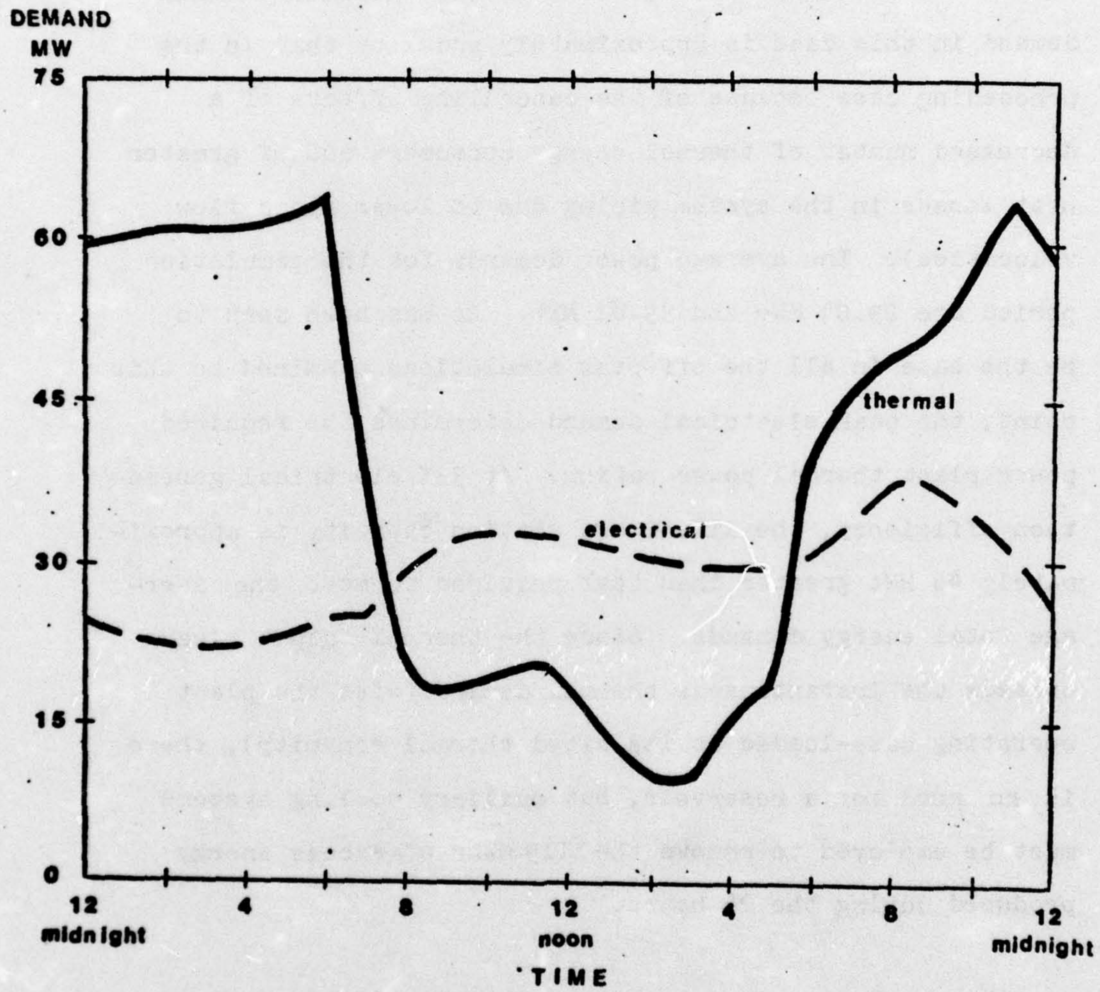


5.1.3.2 75% Thermal/Electrical Load Split Option

The 75% utility system early spring day energy demand schedules are shown in Fig. 5.12. The electrical demand range is 22.08 MWe to 37.79 MWe and the thermal limits are 10.49 MWt and 63.80 MWt. (The minimum thermal demand in this case is approximately equal to that in the preceeding case because of the cancelling effects of a decreased number of thermal energy consumers and of greater heat losses in the system piping due to lower fluid flow velocities). The average power demands for the simulation period are 29.04 MWe and 39.81 MWt. As has been seen to be the case in all the off-peak simulations examined to this point, the peak electrical demand determines the required power plant thermal power rating. At 33% electrical generation efficiency, the 114.52 MWt station capacity is approximately 46 MWt greater than that required to meet the average total energy demands. Since the thermal output always exceeds the instantaneous thermal demand (with the plant operating base-loaded at its rated thermal capacity), there is no need for a reservoir, but auxliary cooling systems must be employed to remove the 1119 MWhr of excess energy produced during the 24 hours.

Figure 5.12

Early Spring Day
75% Thermal/Electrical Load Split
Energy Demand Schedules



5.1.3.3 65% Thermal/Electrical Load Split Option

Figure 5.13 shows the thermal and electrical energy demand schedules computed for the 65% thermal/electrical load split utility system. The minimum electrical load is 24.22 MWe and the maximum is 39.63 MWe; the analogous thermal limits are 4.56 MWt and 54.46 MWt. The average power demands for the 24-hour period are 30.69 MWe and 31.27 MWt. Again, the 120.09 MWt central station capacity is dictated by the peak electrical demand, and the plant is over-rated by approximately 58 MWt for its average total energy demands. No reservoir is required, but auxiliary cooling must dispose of 1424 MWhr of excess generated thermal energy if the plant operates base-loaded at its rated capacity. Since the 120.09 MWt plant is roughly twice as large as that necessary to meet the base's total energy demands on this day, the quantity of energy wasted by operating the plant at a constant total power output would be approximately equal to that used by all the consumers' thermal and electrical equipment.

5.1.4 Late Spring Day Case

The late spring day air temperature schedule shown in Fig. 5.14 is obtained by fitting a smooth curve to the eight 3-hour-interval average temperatures provided from Simmons AFB weather records for the month of May [1]. The

Figure 5.13

Early Spring Day
65% Thermal/Electrical Load Split
Energy Demand Schedules

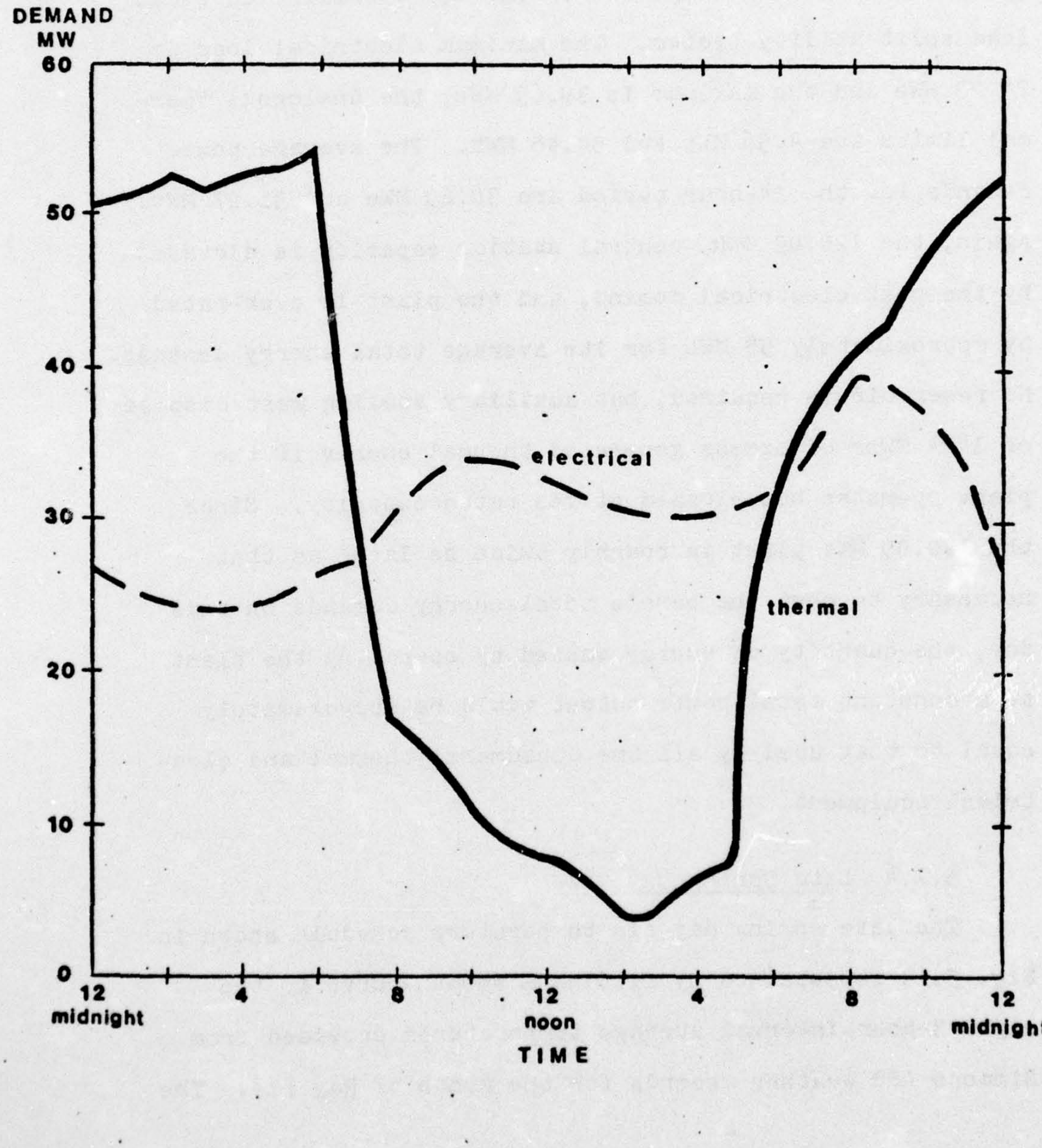
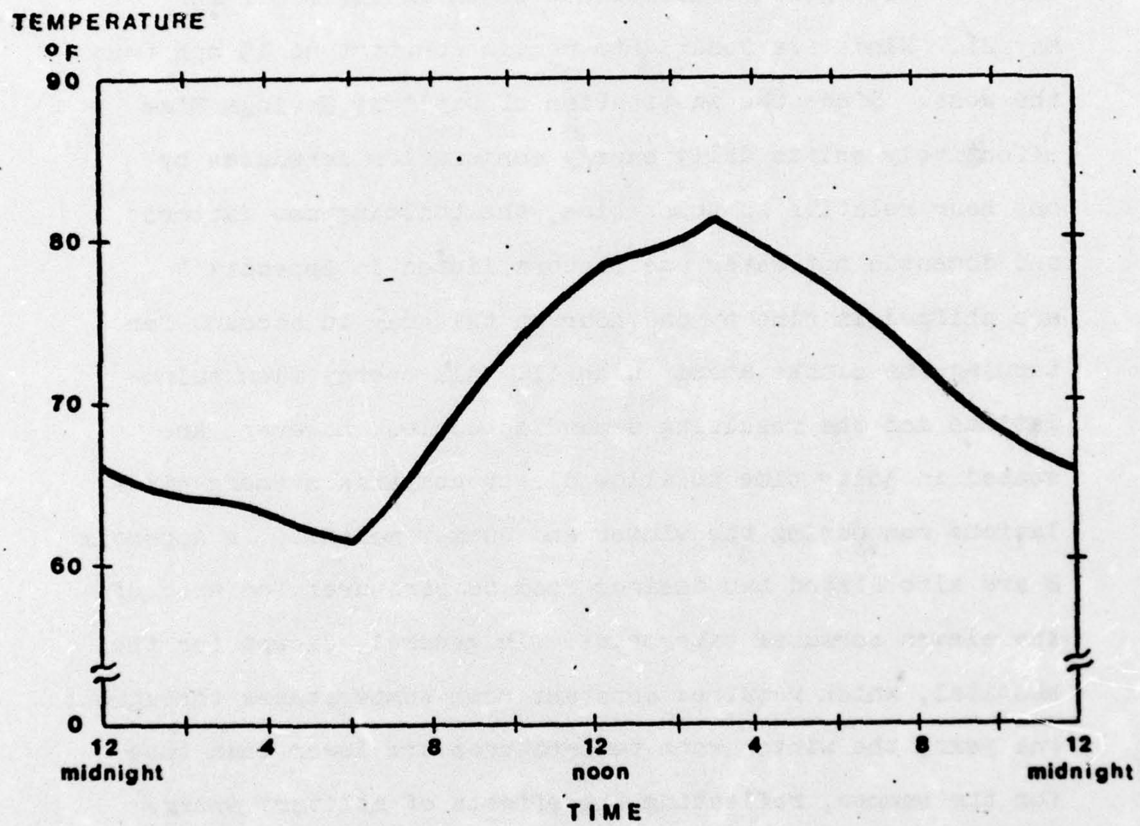


Figure 5.14

Late Spring Day* Air Temperatures

(*Identical to an Early Fall Day)



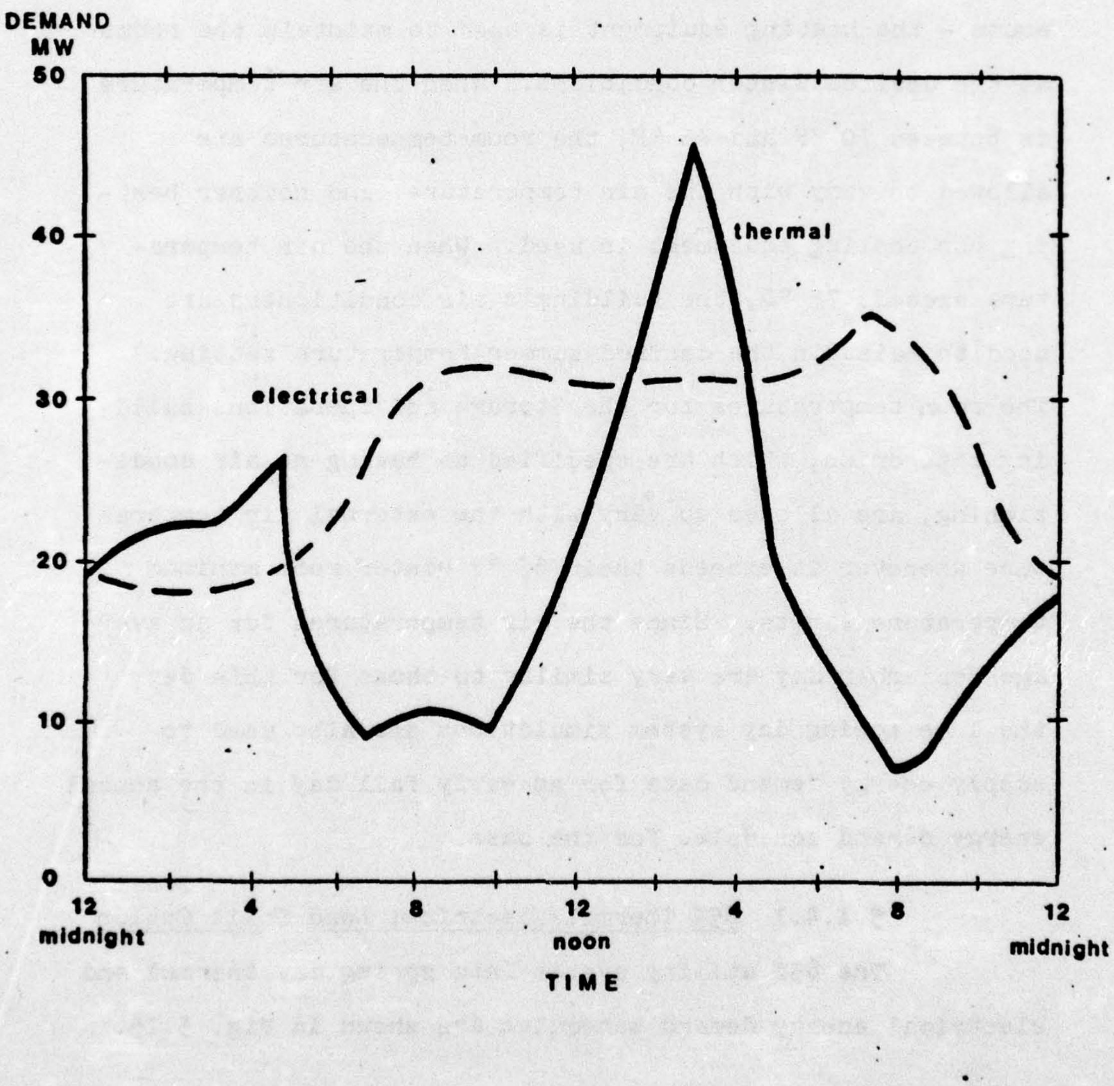
minimum temperature is 61 °F and the maximum is 81 °F. The peak solar radiation intensity of 350 BTU/hr per horizontal square foot of surface area is assumed to obtain under cloudless skies and is modified by the seasonal atmospheric diffusion and absorption coefficients shown in Table 5.2 for May 21. Winds are assumed to remain constant at 15 mph from the west. Since the institution of Daylight Savings Time effectively shifts daily energy consumption schedules by one hour relative to solar time, the building use factors and domestic hot water use factors listed in Appendix B are shifted in time by one hour on this day to account for turning the clocks ahead in April. All energy flow calculations and the resulting demand schedules, however, are stated in solar time to allow direct comparison among simulations run during the winter and summer months. In Appendix B are also listed two desired room temperatures for each of the eleven consumer categories. In general, except for the hospital, which requires constant room temperatures throughout the year, the winter room temperatures are lower than those for the summer, reflecting the effects of military energy conservation practices and generally observed personnel preferences. On this late spring day, when the ambient air temperature lies within the range of the two room temperature settings, the room temperatures are allowed to vary with the air temperature until one of the limits is reached. (e.g., The winter room temperature for the administration building is 70 °F, and its summer setting is 75 °F. When the air temperature is below 70 °F - and solar radiational heating is insufficient to supply the energy lost from the rooms - the heating equipment is used to maintain the rooms at the desired winter conditions. When the air temperature is between 70 °F and 75 °F, the room temperatures are allowed to vary with the air temperature, and neither heating nor cooling equipment is used. When the air temperature exceeds 75 °F, the building's air conditioners are used to maintain the desired summer temperature setting.) The room temperatures for the Storage and Operations building categories, which are specified as having no air conditioning, are allowed to vary with the external air temperature whenever it exceeds their 65 °F winter room minimum temperature limits. Since the air temperatures for an average September day are very similar to those for this day, the late spring day system simulations are also used to supply energy demand data for an early fall day in the annual energy demand schedules for the base.

5.1.4.1 <u>85% Thermal/Electrical Load Split Option</u>

The 85% utility system late spring day thermal and electrical energy demand schedules are shown in Fig. 5.15.

Figure 5.15

Late Spring Day
85% Thermal/Electrical Load Split
Energy Demand Schedules



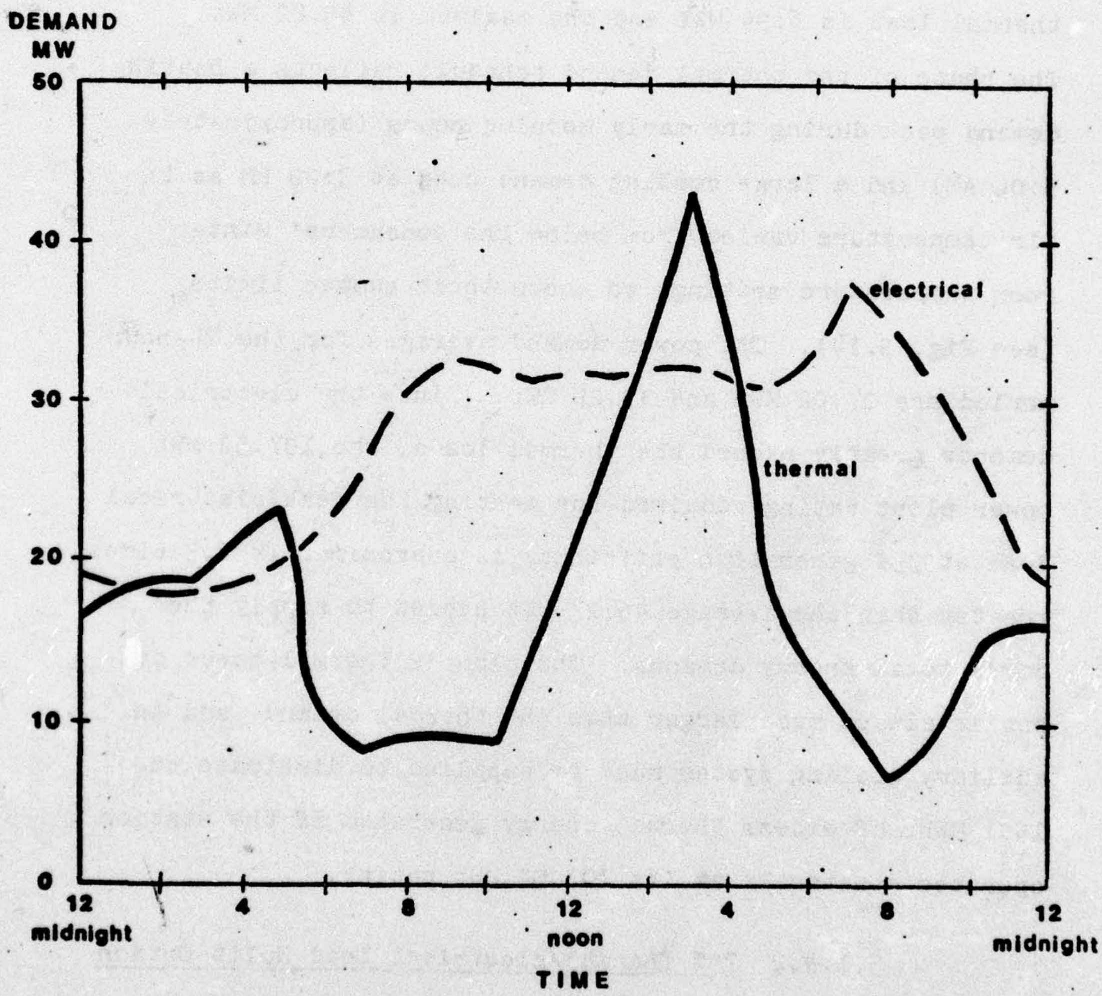
With very little space conditioning load on the system, the minimum electrical demand is 17.86 MWe and the maximum is 35.48 MWe, each of which is only slightly greater than the non-space-conditioning demand limits. The minimum thermal load is 6.94 MWt and the maximum is 45.82 MWt. The shape of the thermal demand schedule reflects a heating demand peak during the early morning hours (approximately 5:00 AM) and a large cooling demand peak at 3:00 PM as the air temperature varies from below the consumers' winter room temperature settings to above their summer limits (see Fig. 5.14). The power demand averages for the 24-hour period are 27.02 MWe and 19.25 MWt. Since the electrical demands greatly exceed the thermal loads, the 107.52 MWt power plant rating required for meeting the peak electrical load at 33% generation efficiency is approximately 2.3 times greater than the average 46.27 MWt needed to supply the day's total energy demands. The plant's thermal power output is always much larger than the thermal demand, and an auxliary cooling system must be supplied to dissipate the 1501 MWhr of excess thermal energy generated if the station operates constantly at its 107.52 MWt rating.

5.1.4.2 75% Thermal/Electrical Load Split Option

Figure 5.16 illustrates the late spring day thermal and electrical energy demand schedules computed for the

Figure 5.16

Late Spring Day
75% Thermal/Electrical Load Split
Energy Demand Schedules



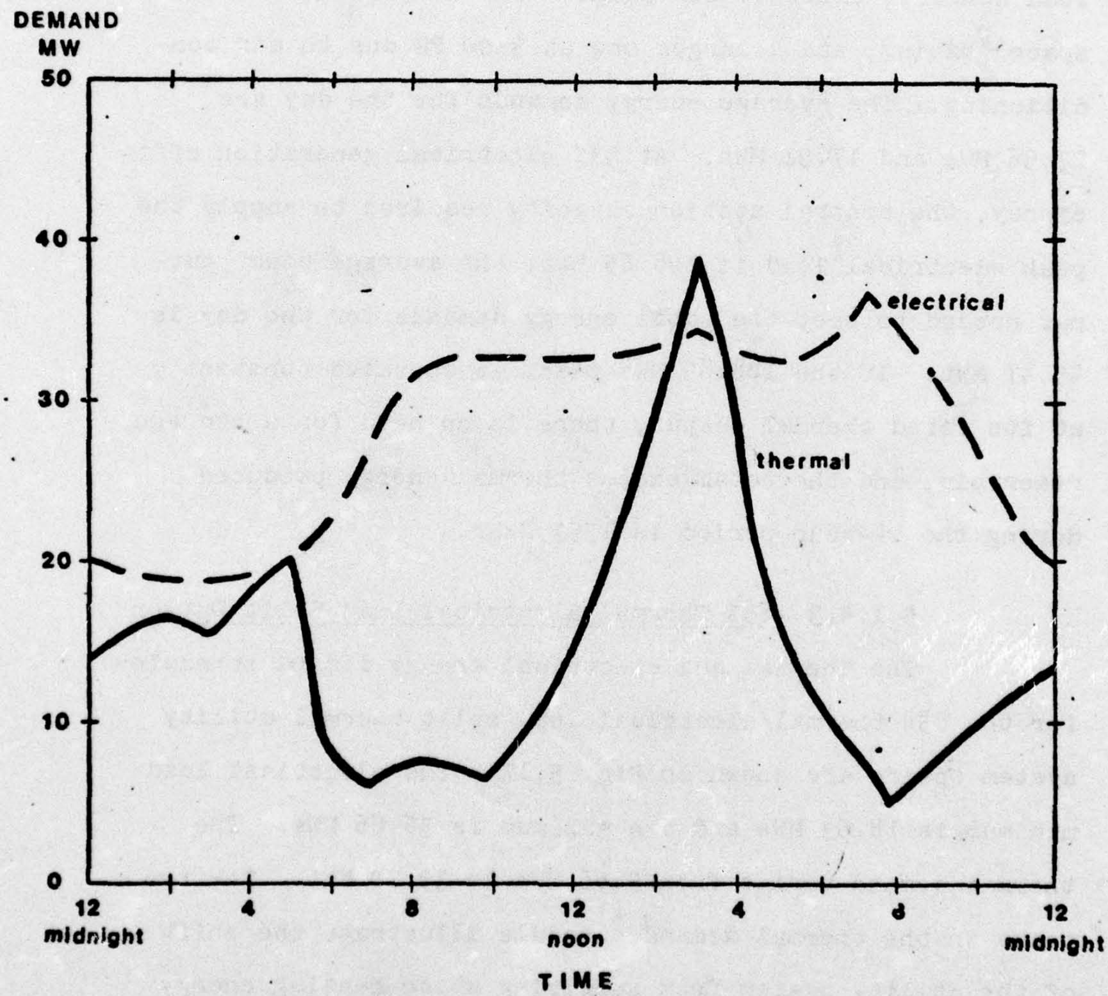
75% thermal/electrical load split thermal utility system option. The electrical load ranges from 18.18 MWe to 35.92 MWe. The thermal demand minimum is 6.81 MWt and the maximum is 43.60 MWt. As in the preceding case, the thermal load schedule exhibits two peaks - one at 5:00 AM due to space heating, and a larger one at 3:00 PM due to air conditioning. The average energy demands for the day are 27.56 MWe and 17.91 MWt. At 33% electrical generation efficiency, the central station capacity required to supply the peak electrical load is 108.85 MWt; the average power output needed to meet the total energy demands for the day is 45.47 MWt. If the 108.85 MWt plant is operated constantly at its rated thermal output, there is no need for a storage reservoir, and the total excess thermal energy produced during the 24-hour period is 1553 MWhr.

5.1.4.3 65% Thermal/Electrical Load Split Option

The thermal and electrical energy demand schedules for the 65% thermal/electrical load split thermal utility system option are shown in Fig. 5.17. The electrical load minimum is 18.63 MWe and the maximum is 36.66 MWe. The thermal demand varies from 5.66 MWt to 39.70 MWt. The two peaks in the thermal demand schedule illustrate the shift of the utility system from supplying space heating energy at night to supplying space cooling energy during the daylight hours. The average power demands for the period shown

Figure 5.17

Late Spring Day
65% Thermal/Electrical Load Split
Energy Demand Schedules



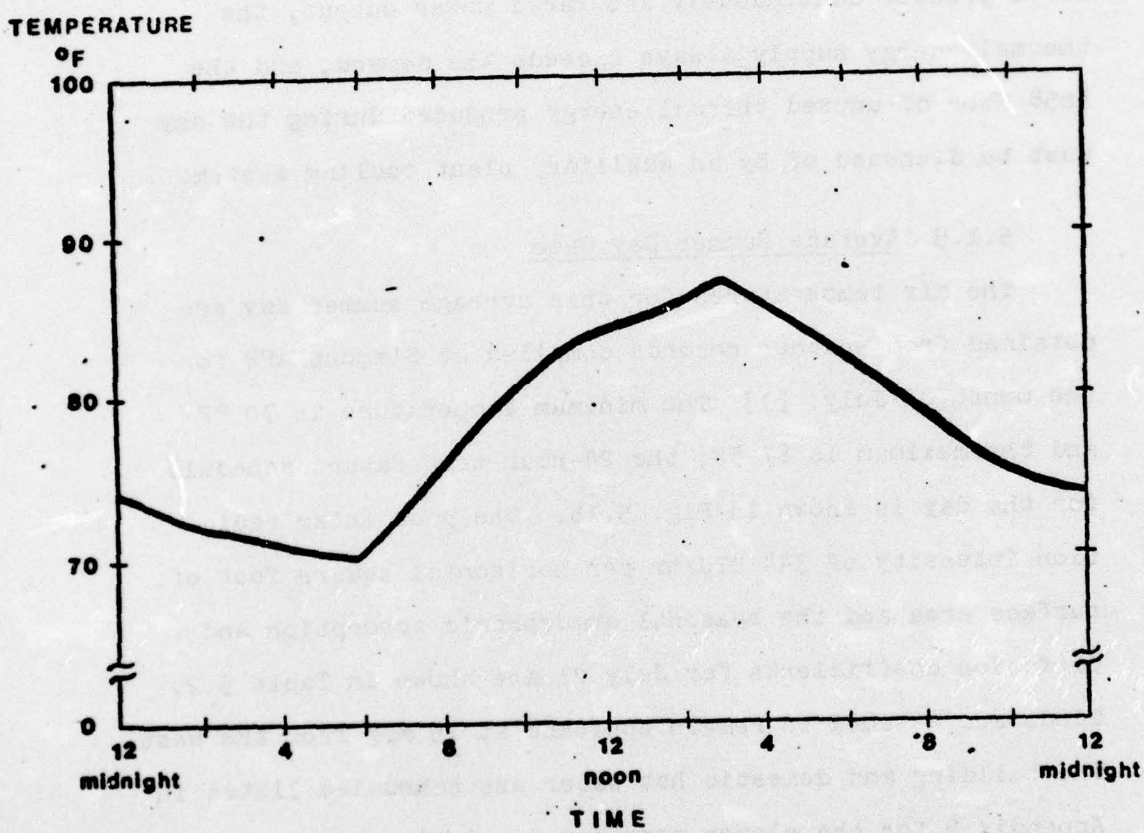
are 28.37 MWe and 15.07 MWt. In order to meet these total energy demands, a power plant capacity of 43.44 MWt is required. However, the 33% generation efficiency limit forces the use of a 111.09 MWt station to supply the peak electrical demand for the day. If this plant operates so as to produce continuously its rated power output, the thermal energy supply always exceeds the demand, and the 1658 MWhr of unused thermal energy produced during the day must be disposed of by an auxiliary plant cooling system.

5.1.5 Average Summer Day Case

The air temperatures for this average summer day are obtained from weather records compiled at Simmons AFB for the month of July. [1] The minimum temperature is 70 °F and the maximum is 87 °F; the 24-hour temperature schedule for the day is shown in Fig. 5.18. The peak solar radiation intensity of 344 BTU/hr per horizontal square foot of surface area and the seasonal atmospheric absorption and diffusion coefficients for July 21 are shown in Table 5.2. Winds are assumed to remain constant at 15 mph from the west. The building and domestic hot water use schedules listed in Appendix B for the eleven consumer categories and the non-space-conditioning electrical load schedule in Table 5.1 are all shifted by one hour in this day's simulations to account for the effects of Daylight Savings Time. As is explained

Figure 5.18

Average Summer Day Air Temperatures

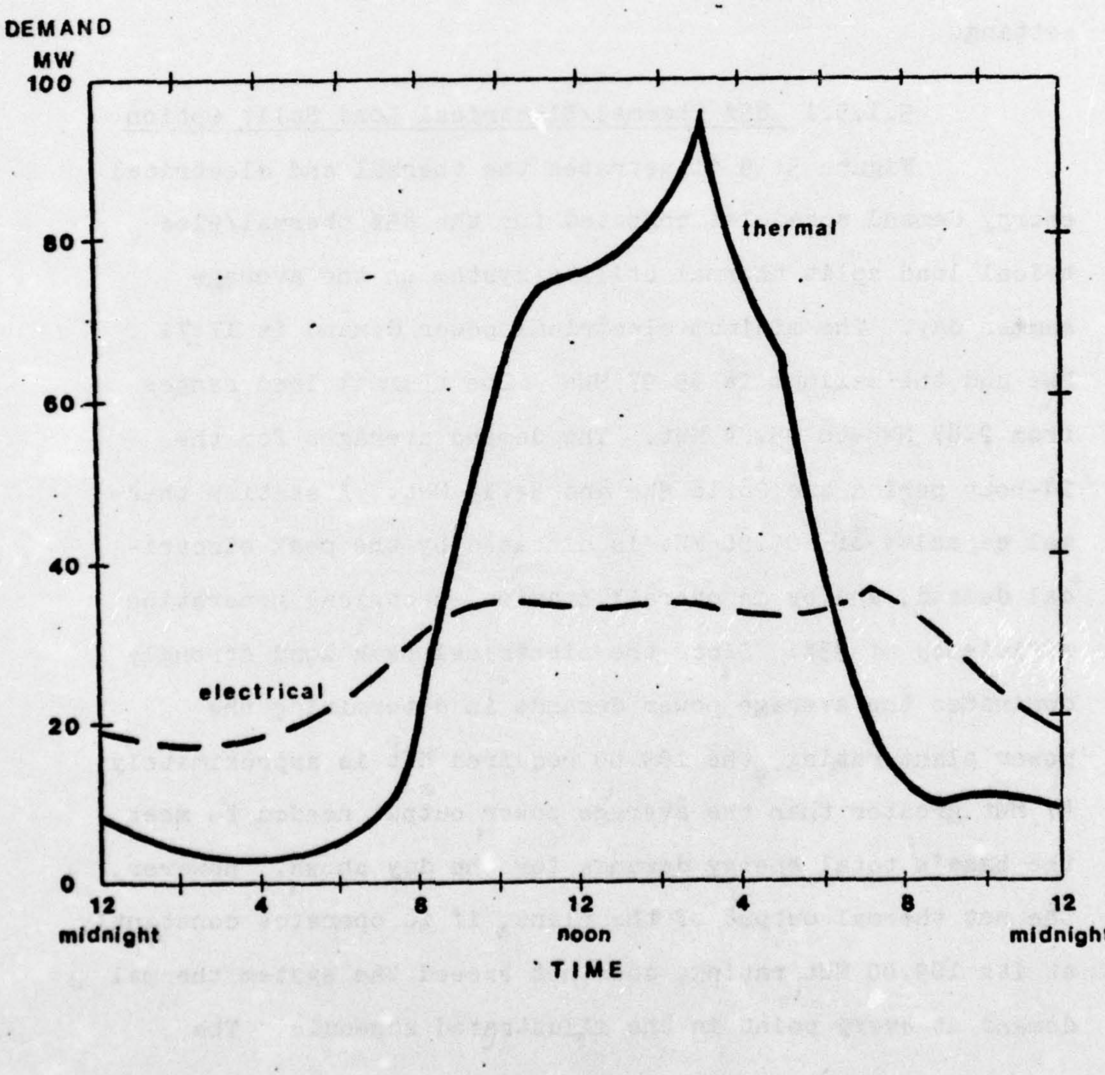


in the description of the late spring day simulations in Section 5.1.4, the building room temperatures are allowed to vary with the air temperatures and no space conditioning equipment is used when the air temperature is between the limits of the buildings' winter and summer temperature settings.

5.1.5.1 <u>85% Thermal/Electrical Load Split Option</u> - Figure 5.19 illustrates the thermal and electrical energy demand schedules computed for the 85% thermal/electrical load split thermal utility system on the average summer day. The minimum electrical power demand is 17.71 MWe and the maximum is 35.97 MWe. The thermal load ranges from 2.87 MWt to 95.84 MWt. The demand averages for the 24-hour period are 28.18 MWe and 34.19 MWt. A station thermal capacity of 109.00 MWt is dictated by the peak electrical demand, and by an overall station electrical generation efficiency of 33%. Since the electrical peak load strongly dominates the average power demands in determining the power plant rating, the 109.00 required MWt is approximately 47 MWt greater than the average power output needed to meet the base's total energy demands for the day shown. However, the net thermal output of the plant, if it operates constantly at its 109.00 MWt rating, does not exceed the system thermal demand at every point in the illustrated schedule. The

Figure 5.19

Average Summer Day
85% Thermal/Electrical Load Split
Energy Demand Schedules



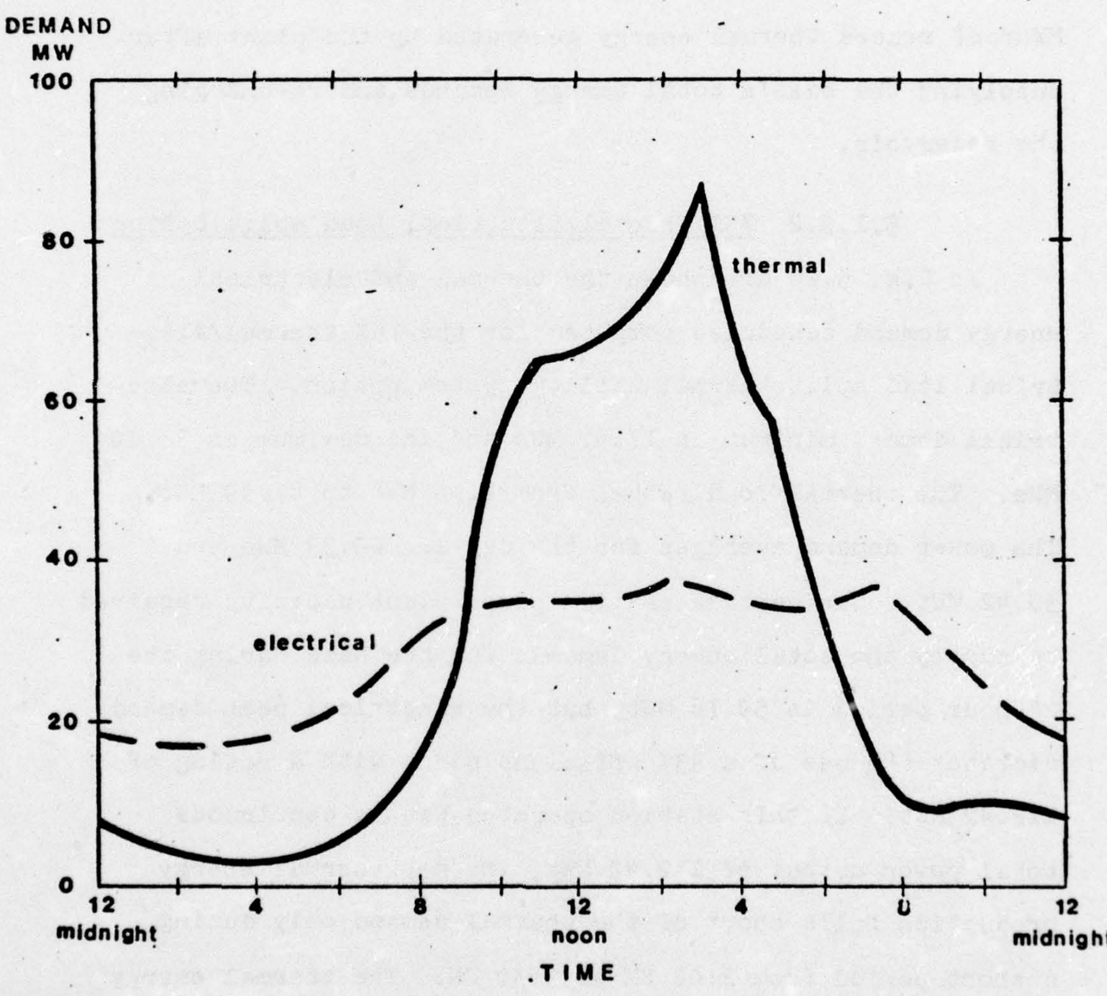
volume of the HTW storage reservoir required to smooth the rather small supply deficit is 6912 cubic feet (61,696 gallons), and the reservoir could be contained within a single cylindrical tank with a height and diameter of 20.65 feet. Auxiliary cooling is required to dissipate the 1142 MWhr of excess thermal energy generated by the plant after supplying the base's total energy demands and re-charging the reservoir.

5.1.5.2 75% Thermal/Electrical Load Split Option

In Fig. 5.20 are shown the thermal and electrical energy demand schedules computed for the 75% thermal/electrical load split thermal utility system option. The electrical demand minimum is 17.92 MWe and the maximum is 37.10 MWe. The thermal load ranges from 2.15 MWt to 87.49 MWt. The power demand averages for the day are 29.33 MWe and 30.42 MWt. The central station power plant capacity required to supply the total energy demands for the base during the 24-hour period is 59.76 MWt, but the electrical peak demand dictates the use of a 33% efficient plant with a rating of 112.42 MWt. If this station operates with a continuous total power output of 112.42 MWt, its net thermal energy production falls short of the thermal demand only during a short period from 2:00 PM to 3:30 PM. The thermal energy

Figure 5.20

Average Summer Day
75% Thermal/Electrical Load Split
Energy Demand Schedules



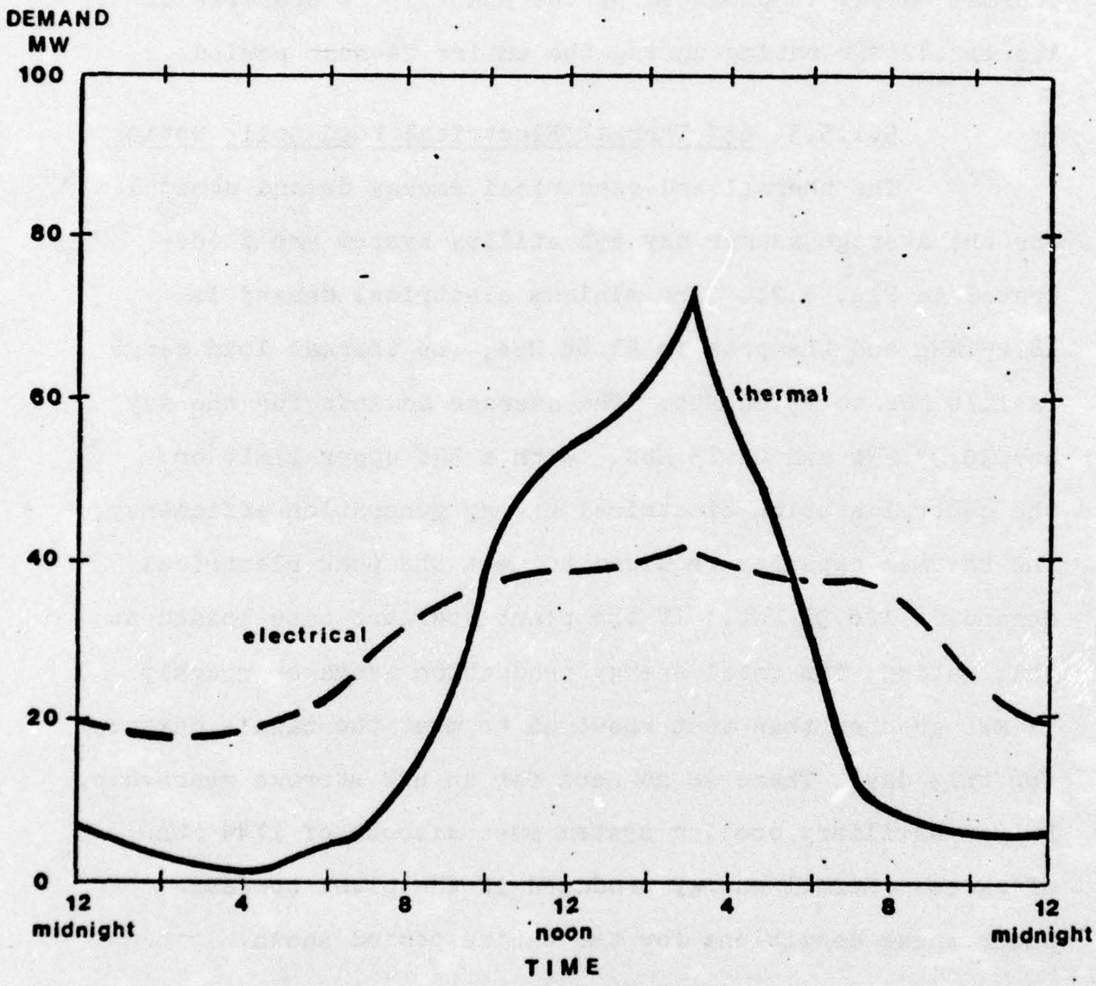
storage reservoir volume required to supply this energy deficit is 2208 cubic feet (16,517 gallons), and the reservoir could be contained within a cylinder having a diameter and height of 14.11 feet. A net excess of 1290 MWhr of thermal energy is produced by the plant if it operates at its 112.42 MWt rating during the entire 24-hour period.

5.1.5.3 65% Thermal/Electrical Load Split Option

The thermal and electrical energy demand schedules for the average summer day 65% utility system are illustrated in Fig. 5.21. The minimum electrical demand is 18.25 MWe and the peak is 41.88 MWe; the thermal load range is 1.70 MWt to 73.66 MWt. The average demands for the day are 30.93 MWe and 24.75 MWt. With a 33% upper limit on the central station electrical energy generation efficiency, the thermal capacity required to meet the peak electrical demand is 126.91 MWt. If the plant operates base-loaded at this rating, its total energy production averages roughly 71 MWt greater than that required to meet the base's demands for this day. There is no need for an HTW storage reservoir, but an auxiliary cooling system must dispose of 1744 MWhr of excess thermal energy produced if the plant operates under these conditions for the entire period shown.

Figure 5.21

Average Summer Day
65% Thermal/Electrical Load Split
Energy Demand Schedules

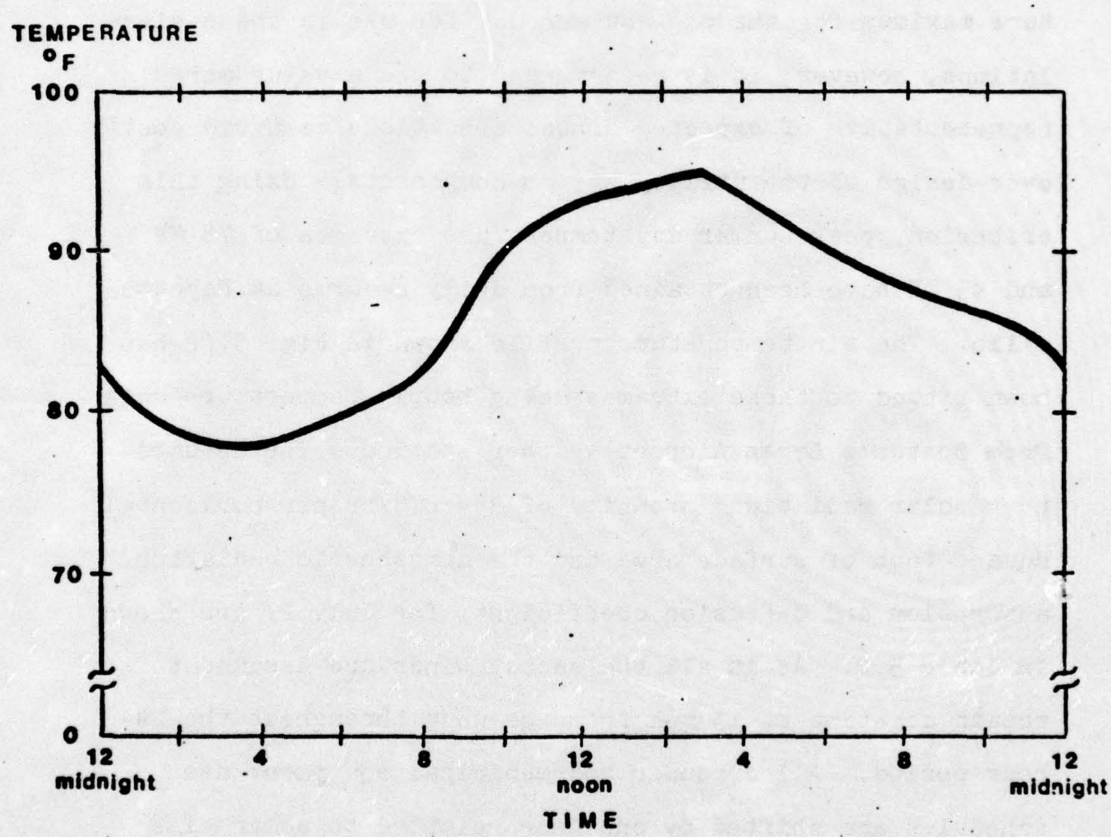


5.1.6 Summer Peak Cooling Demand Day Case

Records from the NOAA weather station at Fayetteville indicate that the highest temperature reached at Ft. Bragg during the past 40 years is 102 °F. In choosing a temperature maximum for the peak summer day for use in these simulations, however, it is recommended to use a value more representative of expected annual conditions to avoid costly over-design of the utility system components. Using this criterion, peak summer day temperature extremes of 78 °F and 95 °F have been obtained from daily records at Fayetteville. The air temperature profile shown in Fig. 5.22 has been fitted to these extremes using hourly temperature data from Boston's Logan Airport weather station. The assumed peak solar radiation intensity of 344 BTU/hr per horizontal square foot of surface area and the atmospheric radiation absorption and diffusion coefficients for July 21 are shown in Table 5.2. As in all the cases, winds are assumed to remain constant at 15 mph from the west throughout the 24-hour period. All consumer and municipal equipment use schedules are shifted by one hour relative to solar time to account for the effects of Daylight Savings Time. Since the 78 °F minimum air temperature exceeds all the desired room temperatures (except, of course, those for the Storage and Operations building categories, which have no air conditioning equipment), the summer day room temperature settings

Figure 5.22

Summer Peak Cooling Demand Day Air Temperatures



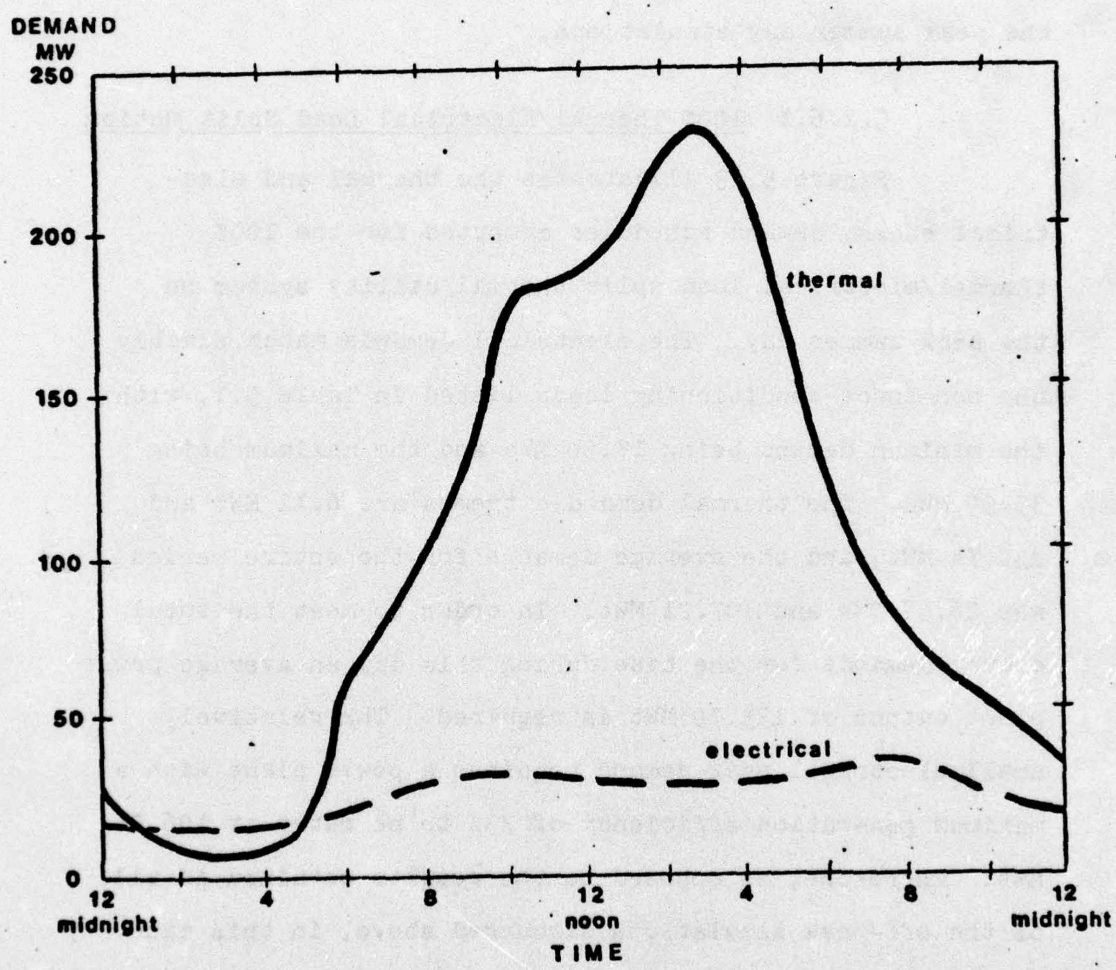
listed in Appendix B apply throughout this day's simulations. To aid in defining a range of equipment sizes for the proposed Ft. Bragg TES, the 100% thermal/electrical load split thermal utility system option is included in the peak summer day simulations.

5.1.6.1 100% Thermal/Electrical Load Split Option

Figure 5.23 illustrates the thermal and electrical energy demand schedules computed for the 100% thermal/electrical load split thermal utility system on the peak summer day. The electrical demands match exactly the non-space-conditioning loads listed in Table 5.1, with the minimum demand being 17.50 MWe and the maximum being 35.00 MWe. The thermal demand extremes are 6.11 MWt and 230.74 MWt, and the average demands for the entire period are 26.58 MWe and 107.21 MWt. In order to meet the total energy demands for the base during this day an average power plant output of 133.79 MWt is required. The relatively small electrical peak demand requires a power plant with a maximum generation efficiency of 33% to be rated at 106.06 MWt. Therefore, as opposed to the results obtained in all of the off-peak simulations discussed above, in this case the average total energy demands dominate the peak electrical load in determining the central station capacity required for use throughout the day. If the power plant operates

Figure 5.23

Peak Summer Day
100% Thermal/Electrical Load Split
Energy Demand Schedules



continuously at its rated 133.79 MWt output, the volume of the HTW storage reservoir needed to smooth the thermal supply and demand imbalances is 235,385 cubic feet (1,760,547 gallons), which could be contained within a cylindrical tank having a diameter and height of 66.92 feet.* Since the 133.79 MWt power plant is sized to be able to meet the base's total energy demands for the day while operating at its rated power output, no excess thermal energy is produced during the period shown.

The primary features of the summertime simulations are illustrated in the results for this case. It is seen that the thermal power demand schedule follows the ascent and descent of the sun in the sky with a phase lag. The thermal demand peak occurs in the late afternoon, and is due almost entirely to the air conditioning load. At night when the air becomes cooled to comfortable temperatures the thermal power demand becomes negligibly small. It is seen also in simulation cases in which there is a significant air conditioning load that the electrical demand schedule displays the type of behavior observed for the 100% thermal/electrical load split thermal load.

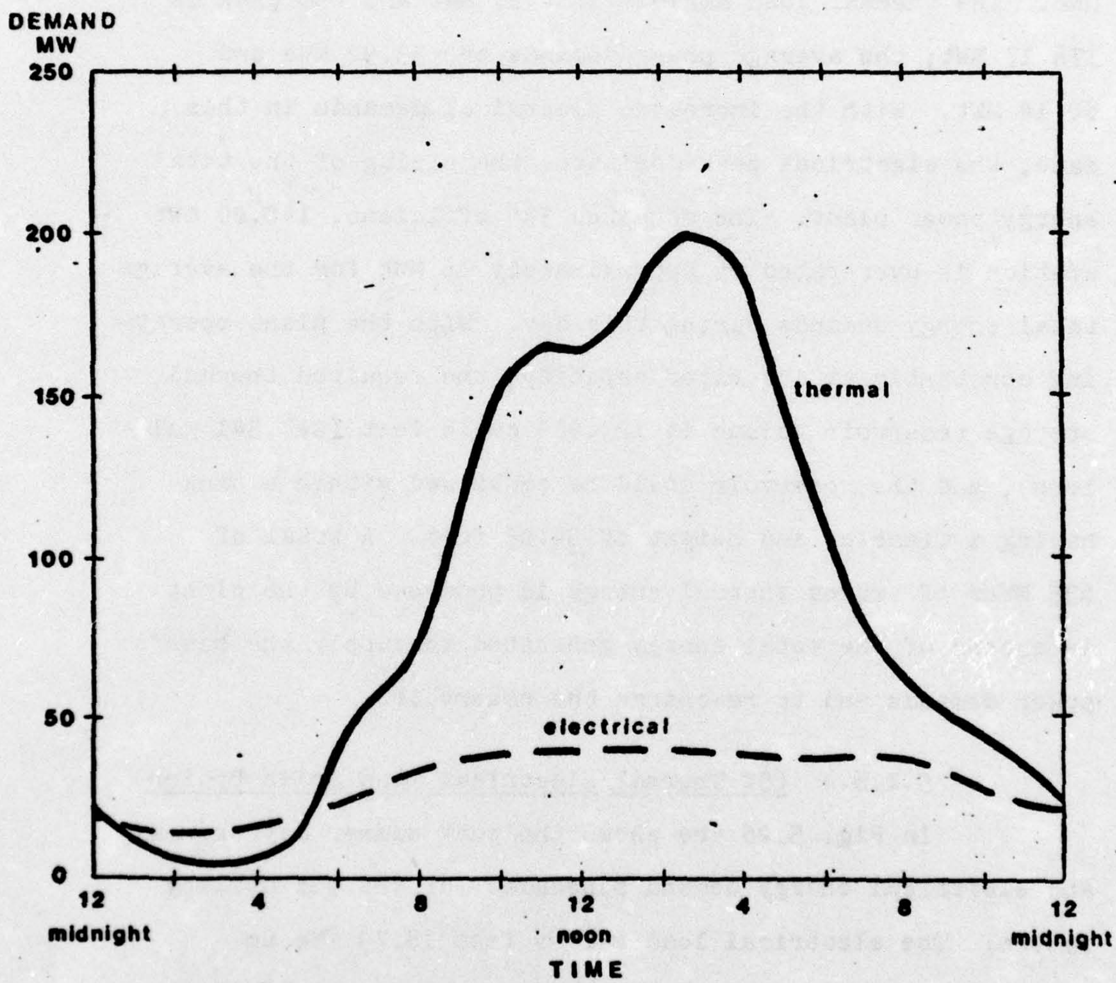
*Again, it must be emphasized that these single tank dimensions are given only for reference and do not imply the use of such a tank in the real system.

5.1.6.2 85% Thermal/Electrical Load Split Option

The peak summer day thermal and electrical energy demand schedules computed for the 85% system option are shown in Fig. 5.24. The electrical power demand minimum is 17.93 MWe and the maximum is 39.58 MWe; the thermal load ranges from 4.98 MWt to 199.95 MWt. The average power demands for the day are 30.89 MWe and 90.29 MWt. To supply the peak electrical load at 33% generation efficiency, the total energy power plant must be rated at 119.94 MWt. A 121.18 MWt plant operating continuously at its rated power output will be able to supply barely the base's total energy demands for the period shown. Although the average demands dominate the station sizing requirements, the difference between the two criteria for this case is only approximately 1.2 MWt, indicating that 121.18 MWt is nearly the minimum plant rating required to supply the base on this day. If this station operates at its rated power output, the HTW reservoir volume required to smooth the thermal supply and demand schedule imbalances is 223,327 cubic feet (1,670,355 gallons), and the reservoir could be contained in a cylindrical tank with a height and diameter of 65.76 feet. No excess thermal energy is produced during the 24-hour period.

Figure 5.24

Peak Summer Day
85% Thermal/Electrical Load Split
Energy Demand Schedules



5.1.6.3 75% Thermal/Electrical Load Split Option

Figure 5.25 shows the thermal and electrical energy demand schedules computed for the 75% thermal/electrical load split thermal utility system option. The minimum electrical demand is 18.38 MWe and the maximum is 46.20 MWe. The thermal load minimum is 4.27 MWt and the peak is 178.17 MWt; the average power demands are 33.92 MWe and 80.14 MWt. With the increased electrical demands in this case, the electrical peak dominates the sizing of the total energy power plant. The required 33% efficient, 140.00 MWt station is over-rated by approximately 26 MWt for the average total energy demands during this day. With the plant operating constantly at its rated capacity, the required thermal storage reservoir volume is 123,985 cubic feet (927,341 gallons), and the reservoir could be contained within a tank having a diameter and height of 54.05 feet. A total of 635 MWhr of excess thermal energy is produced by the plant in excess of the total energy generated to supply the base's power demands and to re-charge the reservoir.

5.1.6.4 65% Thermal/Electrical Load Split Option

In Fig. 5.26 are shown the peak summer day thermal and electrical energy demand schedules for the 65% utility system. The electrical load varies from 18.79 MWe to 56.19 MWe, and the thermal power demand limits are 4.01 MWt

Figure 5.25

Peak Summer Day
75% Thermal/Electrical Load Split
Energy Demand Schedules

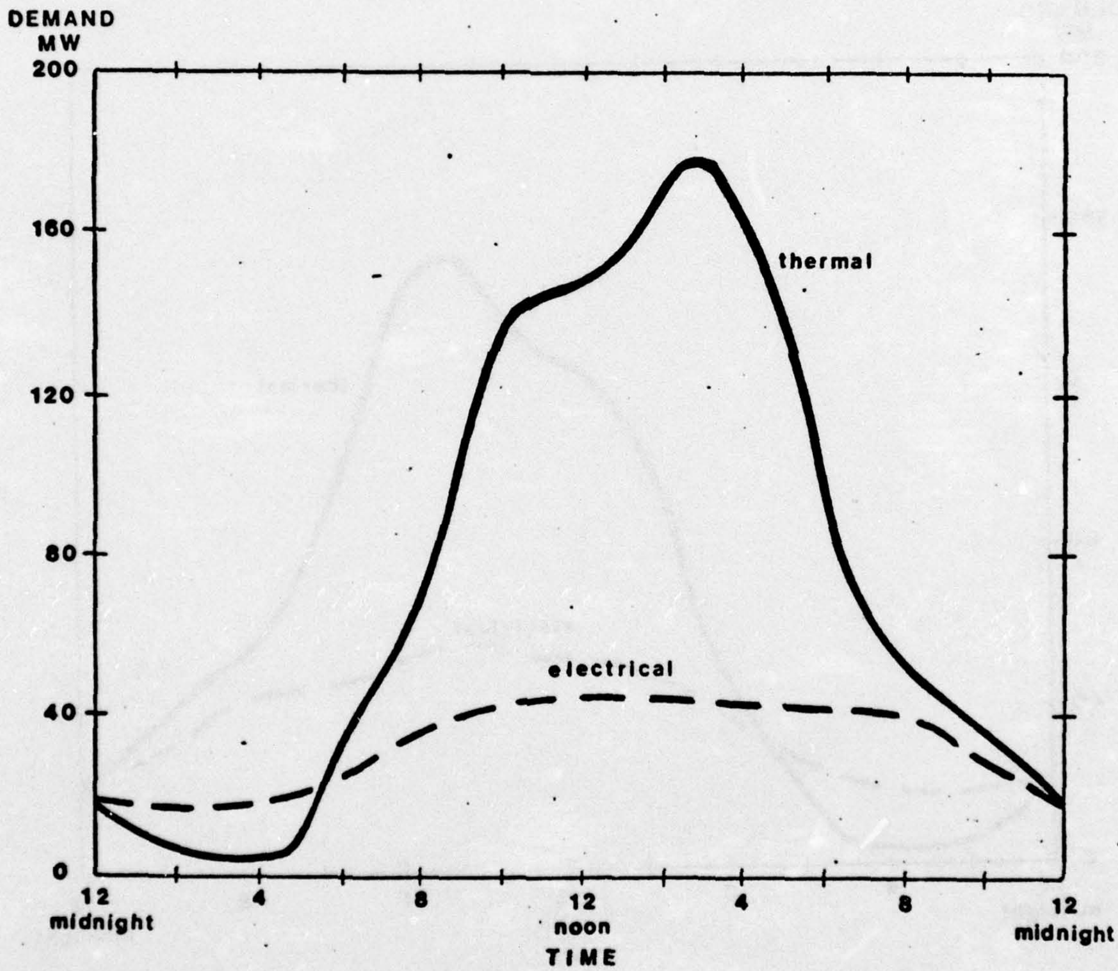
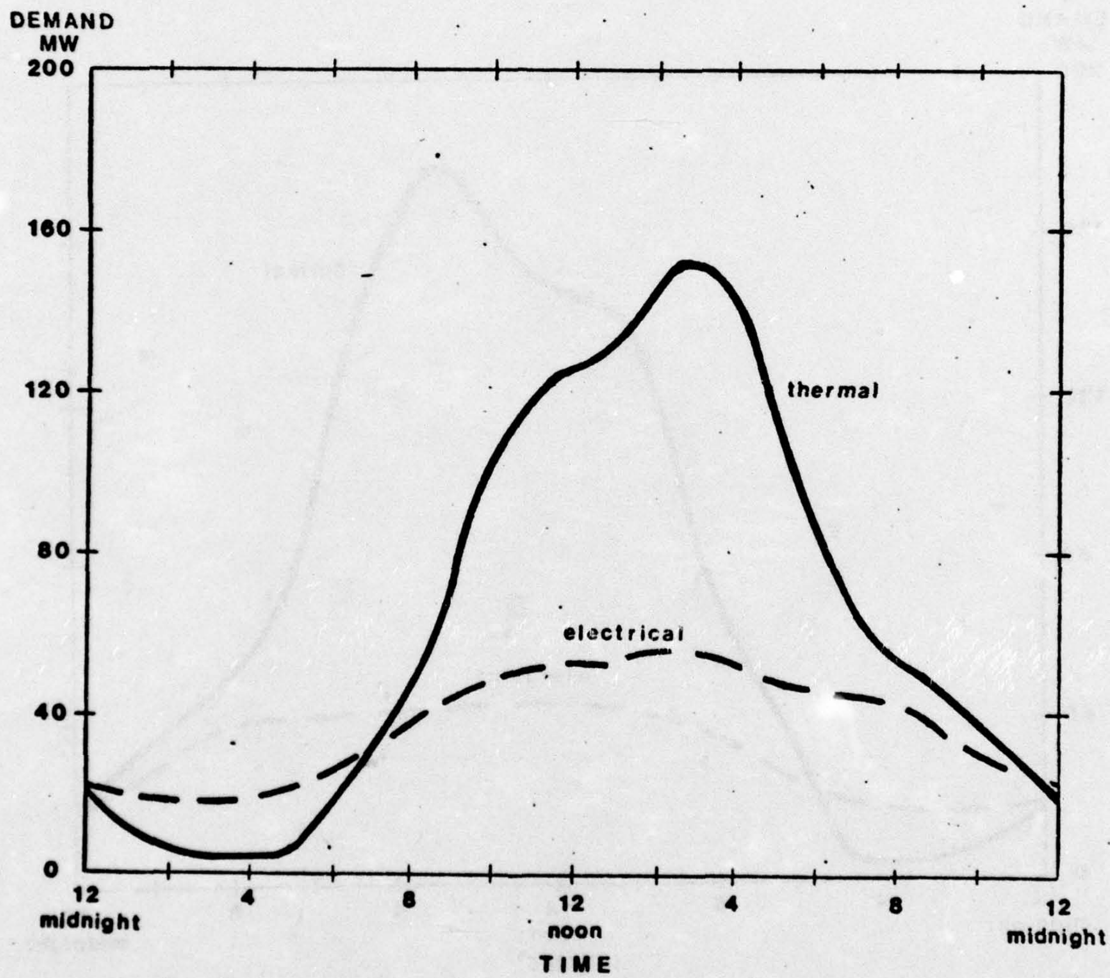


Figure 5.26

Peak Summer Day
65% Thermal/Electrical Load Split
Energy Demand Schedules



and 152.36 MWt. The average 24-hour power demands are 38.10 MWe and 66.95 MWt. In order to meet the peak electrical demand, a 33% efficient central station power plant is required to have a thermal power rating of 170.27 MWt. This capacity is approximately 65 MWt greater than that needed to supply the base's total energy demands for the day shown, but it is not large enough to eliminate the need for an HTW storage reservoir to augment the plant's output during periods of supply deficits. The volume of this reservoir must be 32,332 cubic feet (241,827 gallons), and it could be contained within a cylinder having a diameter and height of 34.53 feet. Auxiliary cooling must also be provided to dissipate the 1598 MWhr of excess thermal energy produced by this station if it operates continuously at its rated 170.27 MWt total power output during the entire 24-hour period.

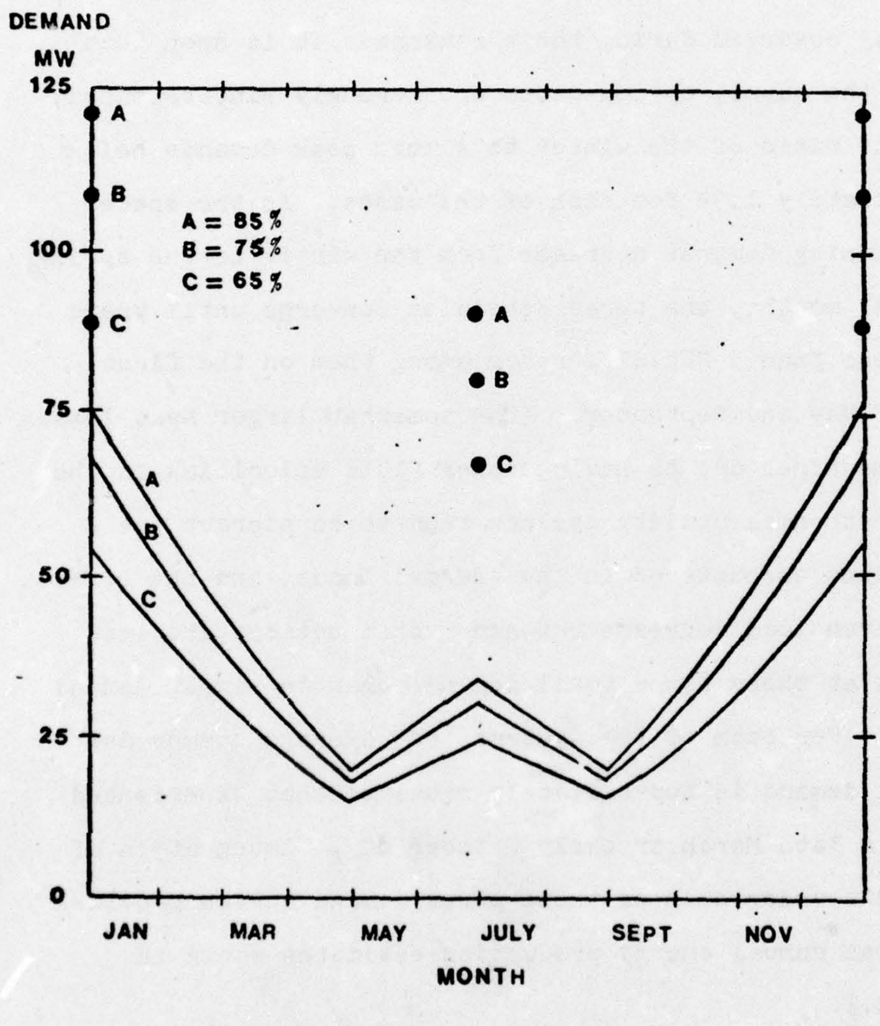
5.2 Annual Energy Demand Schedules

Using the average thermal and electrical energy demands computed in each of the daily simulations described above, it is possible to generate annual demand schedules for Fort Bragg and to determine the total energy consumed during the course of a year for each of the three supply system options studied. (Although the space conditioning, domestic hot water and non-space-conditioning energy demands are determined independently of the energy supply systems used, the total energy consumed in supplying these demands varies with the supply apparatus option chosen due to differences in the average efficiencies of the thermal and electrical end-use equipment employed). In generating these annual demand schedules, it is assumed that the base's energy consumption characteristics are symmetrical between the spring and fall seasons, and the spring day data described in Sections 5.1.3 and 5.1.4 are also used for the autumn days.

Figure 5.27 shows the three annual thermal energy demand schedules for the base. Although the solar radiation data used in each of the simulations is referred to the 21st day of the month, no specific dates have been assigned to any of the other climatological or personnel data, and the average demand data are labeled as corresponding to the first day of each month-studied for easier reference. The three

Figure 5.27

Fort Bragg Annual Thermal Energy Demand Schedules



points plotted above the continuous curves on the first days of January and July are the power demands computed for the peak winter and summer days, respectively, and they represent impulse departures from the conditions normally observed during these seasons. It is seen that all of the supply option cases are strongly winter-peaked, with the ratio of the winter to summer peak demands being approximately 1.34 for each of the cases. As the space conditioning demands decrease from the winter to the spring and fall months, the three schedules converge until there is a less than 5 MWt difference among them on the first days of May and September. (The somewhat larger heat losses from the pipes due to having lower fluid velocities in the smaller thermal utility systems tend to counteract the reductions encountered in the thermal loads, and the effects of a given load decrease between system options are less evident at these lower total demands than in higher demand cases). For each of the systems, the average summer day thermal demand is approximately equal to that experienced during a late March or early October day. Integration of the areas under each of these power demand curves provides the total annual energy production estimates shown in Table 5.3.

Figure 5.28 illustrates the annual electrical energy consumption schedules obtained from the daily average power

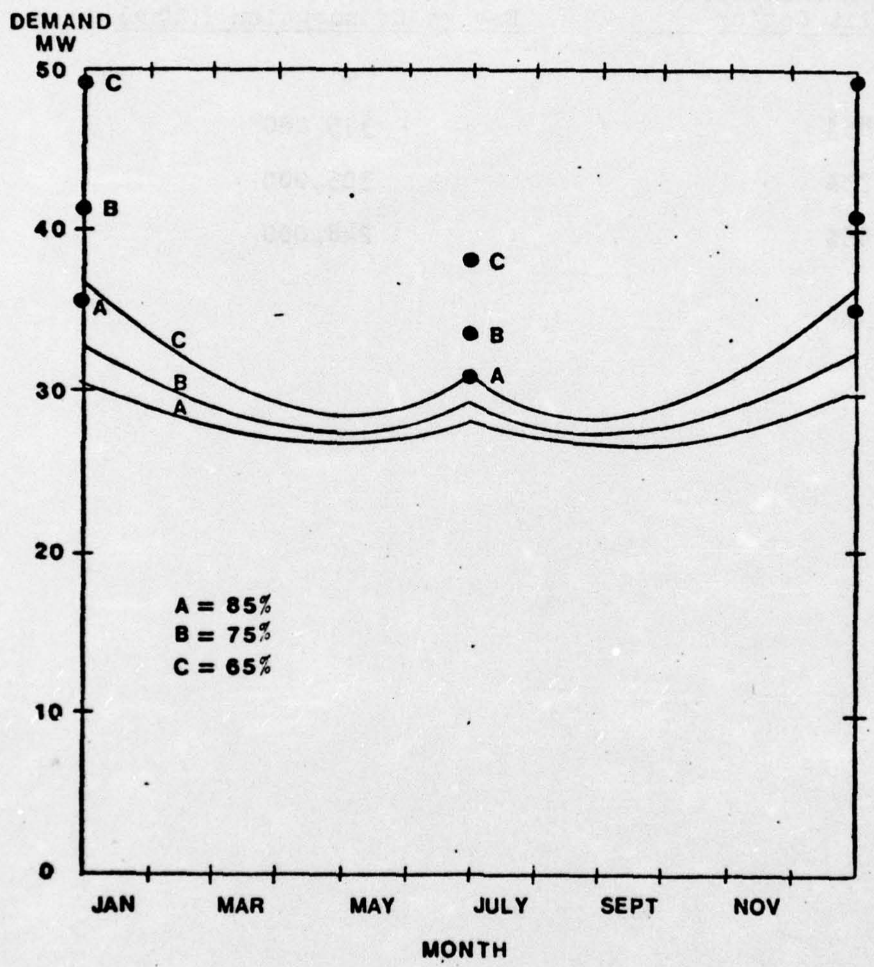
TABLE 5.3

ANNUAL THERMAL ENERGY CONSUMPTION ESTIMATES

Thermal/Electrical Split Option	Energy Consumption (MWhr)
85%	345,000
75%	305,000
65%	248,000

Figure 5.28

Fort Bragg Annual Electrical Energy Demand Schedules



demands presented in Section 5.1. Due to the constant non-space-conditioning load component dominating each of these schedules, they vary less dramatically, and are more nearly equal at their spring and fall minima than are the thermal demand schedules. As is shown in Fig. 5.27, the three independent points plotted for the first days of January and July indicate the peak winter and summer energy demands for the three options, which are impulse departures from the normally observed seasonal loads. Because the electrical loads increase as the thermal supply system size is reduced, the ordering of these curves - with their percentage designations referenced to the nominal thermal supply options - is inverted from that of the thermal demand schedules. Since the winter space heating loads dominate the annual energy demands for both the thermal and the electrical supply systems, the electrical demand schedules are slightly water-peaked and have an average winter-to-summer peak demand ratio of approximately 1.21. (This ratio decreases slightly from the 65% load split system to the 85% load split system due to the larger influence of the annually constant demands in the lower space conditioning load systems). Integrating the areas under these curves yields the total annual electrical energy consumption results summarized in Table 5.4.

TABLE 5.4

ANNUAL ELECTRICAL ENERGY CONSUMPTION ESTIMATES

Thermal/Electrical Split Option	Annual Electrical Energy Consumption (MWhr)
85%	245,000
75%	253,000
65%	270,000

5.3 Central Station Power Plant Rating and Operation Characteristics

The discussions of the power plant ratings required to supply each of the three utility system options during the six daily simulations described in Section 5.1 have been presented in the context of individually sizing a separate plant for each case studied. Of course, each utility system option has only a single central station to supply its total annual energy demands, and that station must be rated so as to be capable of meeting the base's yearly peak loads without exceeding its specified operating characteristics. The mode of operation assumed for each of the power plants during every simulation day is that of constant thermal power production during the 24-hour period. Thus, for example, if the station is rated to produce 100 MWt and the system electrical load varies from 10 MWe to 30 MWe, the electrical output of the plant is assumed to follow the electrical load schedule and the station's net thermal power output ranges from 90 MWt to 70 MWt. In choosing a single station to supply a given utility system's total annual demands, the plant is assumed to operate in this constant energy production mode during short-term demand periods (e.g., on the order of days), but it is not restricted to operate continuously throughout the year at its maximum rated power level. In order to meet the peak winter day's demands, for example, the station might be constructed with a thermal power rating of 200 MWt, and it would operate at this power during the 24-hour period surrounding the system demand peak. During off-peak months, however, the daily system load schedule might require only a maximum of 100 MWt for the plant rating. The station's power output would be reduced to that level, and the plant would operate at a constant 100 MWt output during the short-term daily fluctuations in its demands.

In Figures 5.29-5.31, the annual thermal and electrical energy demand schedules from Figures 5.27 and 5.28 are added directly to produce the annual total energy demand schedules for the base's three utility system options. Also shown in these Figures are the power plant energy production schedules for each of the system options, which differ from the demand schedules by the amount of the excess thermal energy production required for generation of the peak electrical power outputs at a maximum thermal-to-electrical energy conversion efficiency of 33%. As is evident from these power production schedules, the power plant ratings are allowed to vary on a seasonal basis, but on any given day each station operates at a constant thermal power level. The individual points plotted on the first days of January and July indicate the peak winter and summer total energy demands for the systems and the power plants'

Figure 5.29

Fort Bragg
Annual Total Energy Production and Consumption Schedules
85% Thermal/Electrical Load Split Option

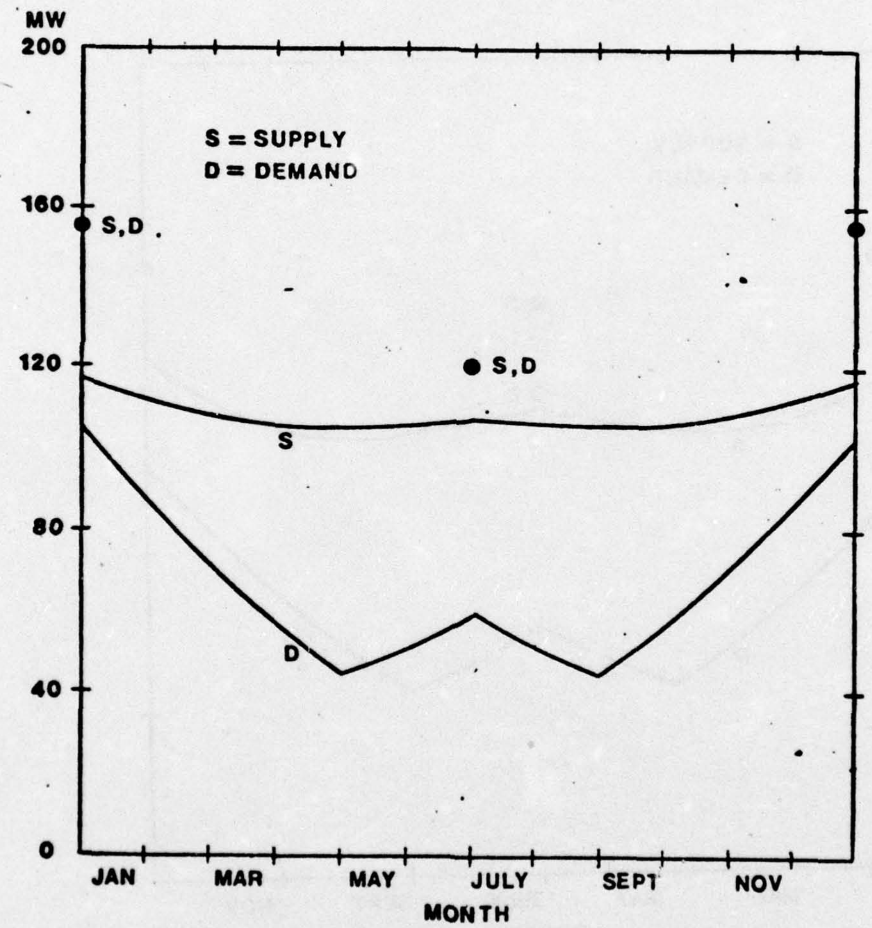


Figure 5.30

Fort Bragg
Annual Total Energy Production and Consumption Schedules
75% Thermal/Electrical Load Split Option

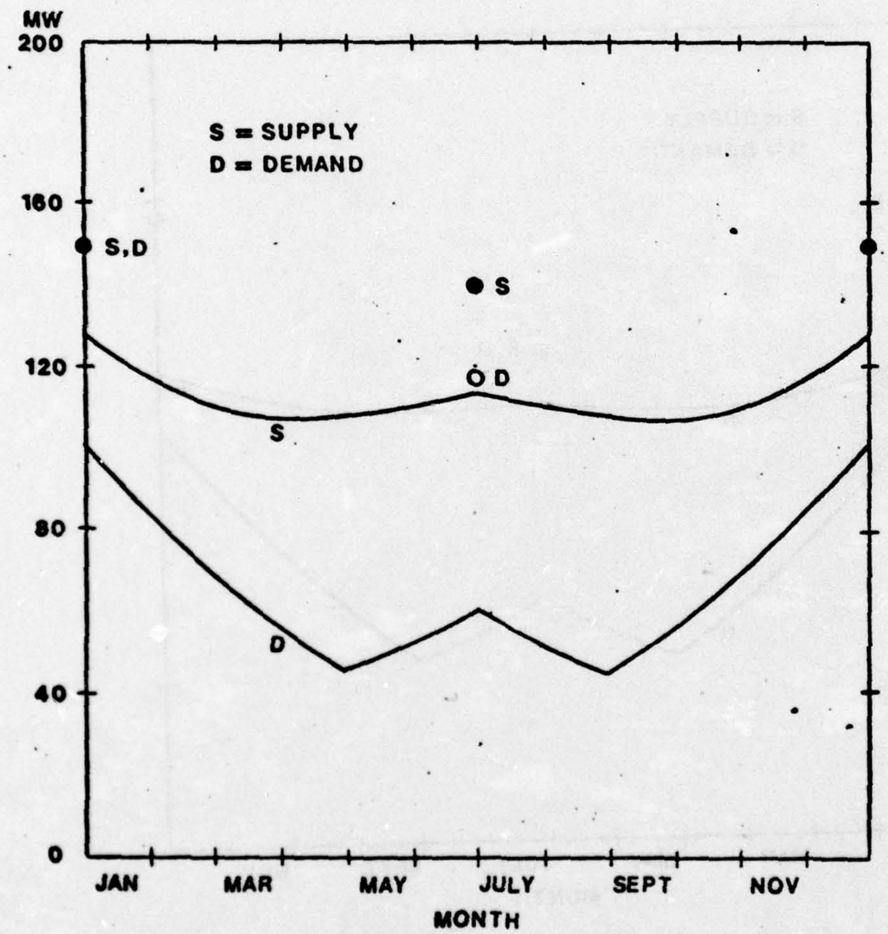
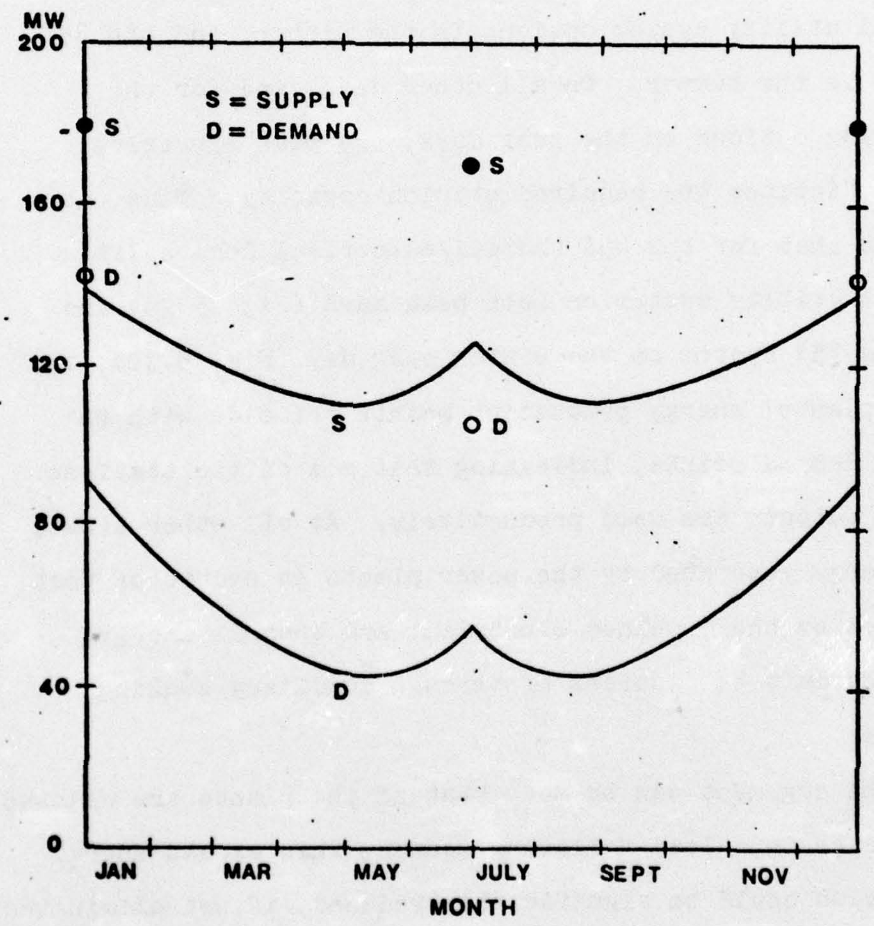


Figure 5.31

Fort Bragg
Annual Total Energy Production and Consumption Schedules
65% Thermal/Electrical Load Split Option



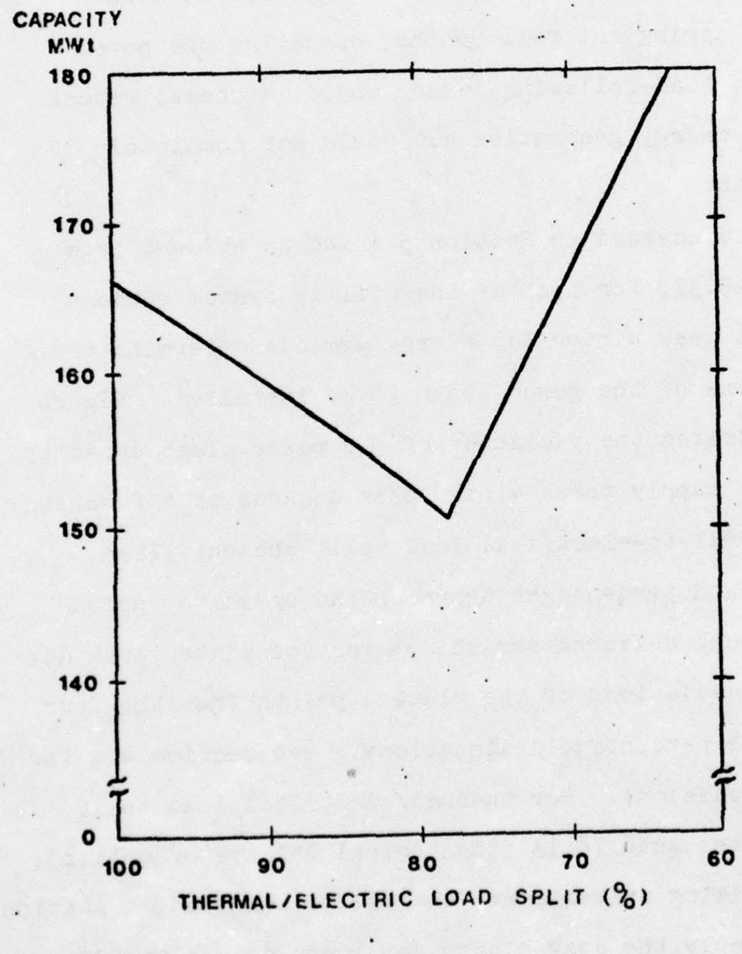
corresponding energy production. It is mentioned in Section 5.1 that the combined thermal and electrical energy demands control the rated power output of the central station only on the two peak-demand days, and only for the 85% and 75% thermal/electrical load split thermal utility system options in the winter, and the 85% option in the summer. On all other days, and for the remaining options on the peak days, the peak electrical demand dictates the required station capacity. Thus, it is seen that for the 85% thermal/electrical load split thermal utility system on both peak days (Fig. 5.29) and for the 75% system on the winter peak day (Fig. 5.30), the power plants' energy production points coincide with the system demand points, indicating that all of the stations' energy outputs are used productively. At all other times, the energy generated by the power plants in excess of that required by the combined electrical and thermal energy networks must be disposed of through auxiliary cooling systems.

The argument can be made that if the plants are allowed to operate in a load-following manner, this excess energy production could be significantly reduced, if not eliminated entirely. Because of the general desirability of operating a large power plant at approximately constant total energy output and because operational variations do not affect the required installed capacity of the plant for each of the options, the load-following operational characteristics of the systems have not been studied in depth. It seems, however, that because of the relatively large electrical demands and relatively small thermal demands at least during the spring and fall months, operating the power plants in a load-following manner would, at best, reduce the excess energy generation but would not completely eliminate it.

As is discussed in Section 5.1 and is evident from Figs. 5.29-5.31, for each of the utility system options studied the peak winter day energy demands determinc the required size of the power plant to be installed. Figure 5.32 illustrates the variation of the power plant capacity required to supply these winter peak demands as a function of the thermal-to-electrical load split chosen. (The thermal demand percentages shown on the ordinate correspond to actual measured conditions for the winter peak day - hence, the deviations of the plotted points from the four nominal case percentage designations. See Section 4.2 for further discussion). For thermal/electrical load split values greater than 78.1% (the nominal 75% system option), the plant sizing criteria are thermally dominated; a station rated to supply the peak electrical power demand at 33%

Figure 5.32

Fort Bragg Total Energy Power Plant
Thermal Rating vs. Thermal/Electrical Demand Split



generation efficiency will be unable to produce enough thermal energy to meet the day's total thermal energy demands. For thermal/electrical load split values less than 78.1%, the system is electrically dominated; a plant operating constantly at the rating required to meet the peak electrical demand is oversized for the day's total energy demands. Only at the split value of 78.1%, and only for the peak winter day, is the station sized such that the capacity required to meet the electrical peak demand matches exactly that required to supply the base's total average thermal energy demands for the 24-hour period. Thus, under the restrictions imposed by the assumed plant operating characteristics, the nominal 75% thermal/electrical load split case affords the minimum installed capacity (150.45 MWt) required for the TES power plant. A more graphic method for explaining the shape of the capacity function shown in Fig. 5.32 is that at thermal/electrical load split values greater than 78.1%, a certain percentage of the power plant's total energy production is thermal energy which is transported directly to the consumers. If 3 units of this excess thermal energy production capacity are used to generate 1 unit of electrical energy for use in heat pumps having an average COP of 2.4, where there were formerly 3 units of heat provided thermally to the consumer, there are now 3-1 = 2 units of turbine exhaust heat produced thermally and 1x2.4 = 2.4

units of heat pump energy produced electrically. Thus, the 3 units of power station thermal energy production can be made to produce 4.4 units of useful heat rather than the three units which had been produced formerly. Conversely, the plant thermal power capacity could be reduced by means of such an end-use apparatus substitution while maintaining the same total energy supply capability. Once the minimum thermal capacity rating is reached, however, all the station's thermal energy production will be used in the generation of electricity (with all of the waste heat being transported to thermal energy consumers). In this case a further increase in the plant's electrical output of 1 electrical unit would require an increase in its thermal rating of 3 units, 2 of which must be disposed of as waste heat from the electrical generation process since thermal energy demand for them would not exist.

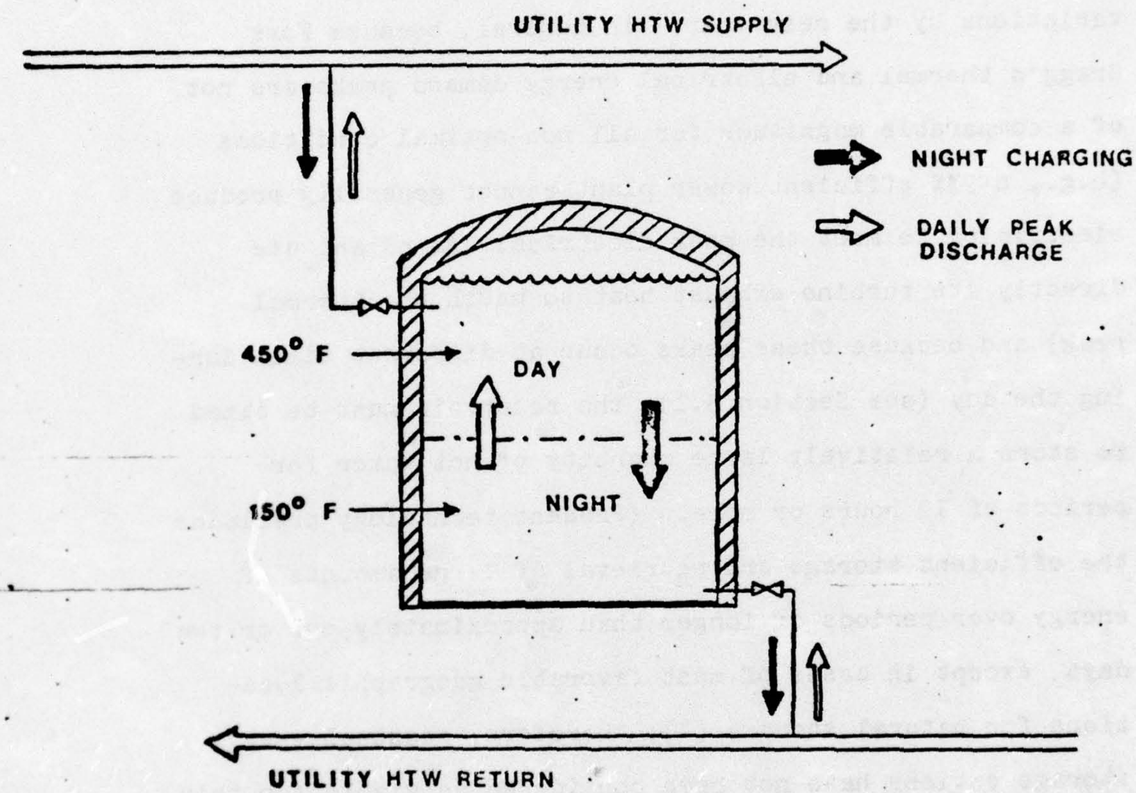
5.4 Sizing of the Thermal Energy Storage Reservoir

The primary function of the thermal energy storage reservoir is to supply the thermal utility system power demands during periods of insufficient power plant output. Its size therefore depends critically upon the assumed mode of power plant operation and upon the thermal energy supply and demand imbalances determined by the variations in the thermal and electrical load schedules. In the preceding section a constant total energy output mode of plant operation has been described, in which the station's electrical power output follows its electrical demand schedule, and its thermal output is buffered from the thermal demand variations by the reservoir. In general, because Fort Bragg's thermal and electrical energy demand peaks are not of a comparable magnitude for all non-optimal conditions (e.g., a 33% efficient power plant cannot generally produce electricity to meet the peak electrical demand and use directly its turbine exhaust heat to match the thermal peak) and because these peaks occur at different times during the day (see Section 5.1), the reservoir must be sized to store a relatively large quantity of hot water for periods of 12 hours or more. (Present technology precludes the efficient storage and retrieval of large amounts of energy over periods of longer than approximately one or two days, except in cases of most favorable geographic locations for natural storage [4]; therefore, seasonal energy storage options have not been considered as viable for this 1985 utility system.)

Figure 5.33 illustrates the proposed underground thermally stratified HTW storage reservoir design. European experience with this method of energy storage, in which the hot/cold water interface rises and falls as the reservoir discharges and is charged with hot water, indicates that mixing of the two temperature regions is minimal for reservoir charge/discharge times on the order of

Figure 5.33

Underground Stratified Thermal Energy Storage Reservoir



12 hours or longer [4] and that, with proper insulation, heat conduction to the surroundings is relatively small. Since the proposed TES uses water at a temperature of 380 °F as the primary supply to the thermal utility system, the reservoir must be pressurized to roughly 250 psia to prevent the HTW's flashing to steam. To date, cylindrical steel storage tanks capable of withstanding this pressure have been limited to a size of roughly 20 feet in diameter and 70 feet long [5]. From Section 5.1, it is seen that the 21,991 cubic-foot volume of one of these tanks does not provide enough capacity to smooth the thermal energy supply and demand imbalances on the peak summer day for any of the three system options. Therefore, the proposed storage reservoir is not to be a single cylindrical tank as indicated by the dimensions referenced in Section 5.1, but it is rather composed of a series of these smaller tanks piped in parallel to provide the required storage volume and a uniform thermal stratification throughout the storage field.

Although the daily simulation descriptions in Section 5.1 present reservoir sizing data in the context of systems designed individually for each of the six days studied, only one day out of the year actually governs the size of the reservoir to be installed for each of the three utility system options. Since the reservoir volume is determined by the maximum discrepancy between the thermal energy supply and demand schedules and not by the absolute magnitudes of these energy flows, the primary criterion to be met by the storage system is that on the most severely imbalanced day of the year, the reservoir, in combination with the thermal energy output of the power plant, must supply enough thermal energy to just meet the utility system demands without being completely discharged and must be fully re-charged during the 12-hour period following the maximum mismatch (to be prepared for the next day's cycle). From Section 5.1 it is seen that, although the day with the maximum total energy consumption is the peak winter design day, the peak summer day's thermal demand schedules exhibit the greatest variations, and therefore, the energy supply and demand conditions on that day determine the reservoir size required for each system option. Table 5.5 presents the sizing criteria and the reservoirs chosen for each of the utility system options as dictated by these summer peak requirements. If the reservoirs are sized to just meet the design volume criterion shown, they will be completely discharged only during the peak summer day, and the position of the hot/cold water interface will vary much less on the remainder of the days throughout the year. In fact, because the 65% thermal/ electrical load split utility system option power plant

TABLE 5.5

THERMAL ENERGY STORAGE RESERVOIR SIZING

Thermal/ Electrical Load Split Option(1)	Water Temperature (°F) Hot	Water Temperature (°F) Cold	Energy Storage (MWhr)	Volume (cubic feet)	Volume (gallons)	Number of Standard Tanks(2) Required	Number of Standard Tanks(2) Installed
85%	380	172	827	223,000	1,670,000	10.14	11
75%	380	151	509	124,000	927,000	5.64	6
65%	380	163	125	32,000	242,000	1.47	2

(1) See Chapter 4.

(2) A standard tank is defined here as a right circular cylindrical steel tank constructed to withstand a working pressure of 250 psia, insulated and installed underground, having a diameter of 20 feet and a length of 70 feet [5].

rating is so strongly dominated by the base's electrical energy demands, that plant's thermal power output exceeds the instantaneous thermal demand at all times except from 11:00 AM to 4:30 PM on the peak summer day, and that 5-1/2 hour period is the only time during the year in which any power plant supplementation is required; the 65% load split utility system reservoir remains fully charged and unused during the other 364 days. The 85% and 75% thermal/electrical load split option reservoirs vary in their use from zero in the spring and fall to roughly 33% and 53% of their respective capacities on the peak winter day. (It should be noted that because of the use of an integral number of standard steel tanks to supply the required storage volume, none of the three reservoirs are cycled completely on any of the simulation days described.)

5.5 Sizing of the Auxiliary Cooling System

The constant thermal power output mode of power plant operation has been shown to provide more thermal power than can be consumed by the Fort's utility system over a wide range of weather conditions for each of the three system options studied. Since it is impractical and uneconomical to size the reservoir to absorb these large quantities of excess energy, some form of auxiliary cooling system must be provided to dissipate the heat produced during periods of high electrical and low thermal power demands. (As is mentioned in Section 5.3, operation of the power plant in a load-following manner has not been investigated, but it is surmised that this operational variation would only slightly reduce the required cooling system size, since even a load-following station would produce large amounts of excess heat during the spring and fall months). This conclusion has been reached following the examination of the daily simulation results, but no attempt has been made to incorporate a dynamic auxiliary cooling system model into the utility system models or to investigate its effects, if any, upon the overall TES performance and transient stability.

Since no detailed cooling system analyses have been performed, no recommendations are made as to the optimal form of this system to interface with each of the utility system options. Due to general environmental protection considerations and due to the relative scarcity of large quantities of water near the Fort Bragg site, however, it is assumed that some form of natural or mechanical draft cooling tower will be constructed rather than using once-through cooling, spray ponds, or some other single or combined system configuration.

Since the dynamic interactions among the central station power plant, the thermal utility system and the thermal energy storage reservoir are computed internally to the

TDIST simulations for each case studied, it is difficult to perform a time-dependent system energy flow analysis based only upon the simulation output data which can be used to accurately determine the point of maximum excess thermal energy production for each of the three systems. (e.g., Although the thermal energy supply may exceed the consumer's demands, some of the surplus thermal energy may be used to recharge the depleted storage reservoir and would not need to be handled by auxiliary cooling towers.) Since for each of the systems the maximum total excess energy generated during a 24-hour period occurs on a day when the reservoir remains unused, the instantaneous peak waste heat generation on these days is used to determine the size of the cooling tower to be installed. Table 5.6 lists the cooling tower sizing data for each of the three system options studied. It should be noted that the proposed cooling towers are sized to dissipate only the maximum waste heat generation expected under normal operating conditions for each of the three TES options; if it is desired to rate the towers to absorb the full output of each of the stations (in case of emergency system shutdown, etc.), the three towers must be sized according to the plant ratings presented in Section 5.3: 85%-156.88 MWt, 75%-150.45 MWt, and 65%-179.03 MWt.

TABLE 5.6

AUXILIARY COOLING TOWER(1) SIZING DATA

Thermal/ Electrical Load Split Option(2)	Day(3)	Time	Water Temperature(°F)	Hourly Peak Excess Heat Production (MWt)	Daily Peak Excess Heat Production (MWhr)	Annual Excess Heat Production (MWhr)
85%	L.S.	6 AM	167	73.38(4)	1501	383,000(7)
75%	L.S.	6 AM	159	75.86(5)	1553	412,000(8)
65%	A.S.	3 PM	138	106.35(6)	1744	538,000(9)

(1) Cooling towers assumed to be used, but the data could be applied to any cooling system.

(2) See Chapter 4.

(3) L.S. = Late Spring; A.S. = Average Summer

(4) The peak hourly production for the A.S. day is 88.12 MWt, but it is partially used to re-charge the reservoir.

(5) The peak hourly production for the A.S. day is 95.32 MWt, but it is partially used to re-charge the reservoir.

(6) The 65% system reservoir is not used on the A.S. day.

(7) See Fig. 5.29.

(8) See Fig. 5.30.

(9) See Fig. 5.31.

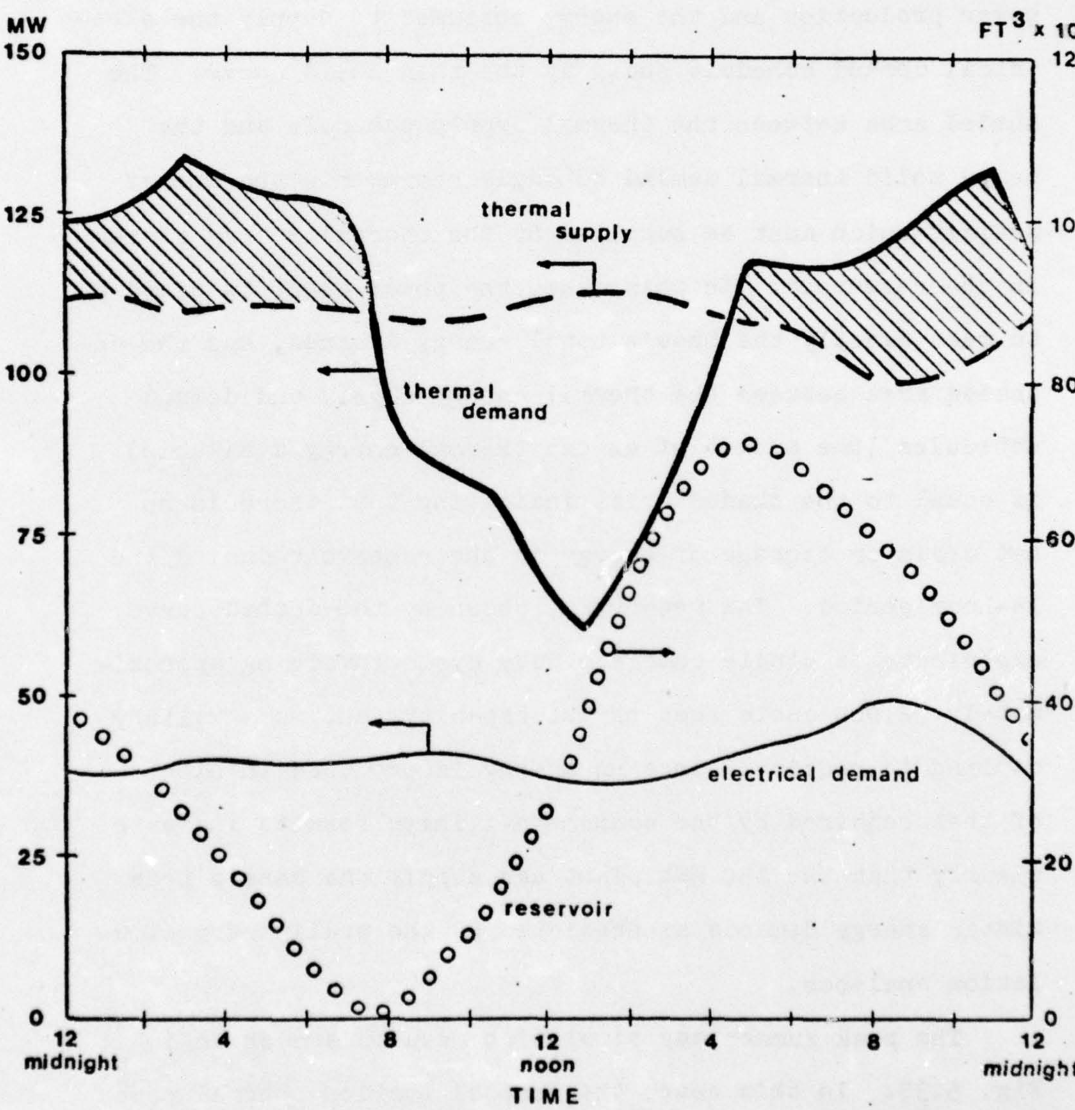
5.6 Verification of System Designs

The power plant ratings, the storage reservoir volumes, and the auxiliary cooling tower capacities computed for the three supply system options are based upon the daily thermal and electrical energy demand data obtained from the simulations described in Section 5.1. However, because these simulations are performed with no foreknowledge of the precise supply system parameters to be used in each case, they provide only rough estimates of the performance of the power plant and reservoir actually planned for the thermal utility system. Therefore, in order to verify the validity of the design parameters chosen for the energy supply components and to insure the stability of their interactions with the thermal and electrical energy distribution networks, a second series of simulations must be performed for at least the peak days governing the design of these components. As an example of this design verification process, summer and winter day simulations have been re-run for the 75% thermal/electrical load split utility system option, which promises to provide the minimum installed power plant capacity necessary to supply the base's annual total energy demands.

Figure 5.34 illustrates the results of the peak winter day simulation with the power plant's thermal power output

Figure 5.34

75% Thermal/Electrical Load Split
Utility System Design Verification
Peak Winter Day Simulation

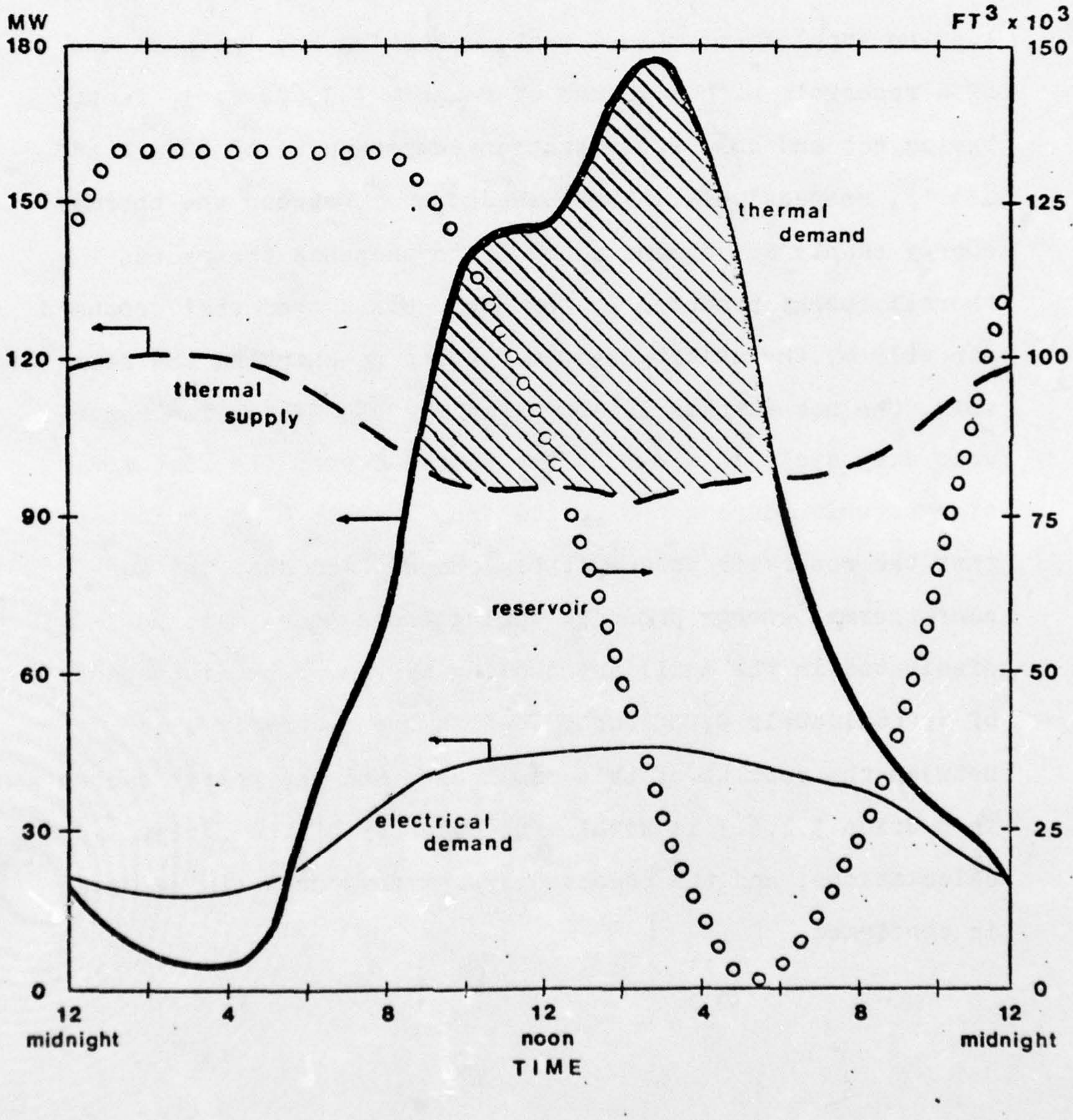


set of the 150 MWt required to just meet the base's total energy demands for the 24-hour period. The dashed line, representing the power plant's thermal energy supply schedule, is simply the difference between the constant 150 MWt power production and the energy consumed to supply the electrical demand schedule shown by the thin solid curve. The shaded area between the thermal supply schedule and the heavy solid thermal demand schedule represents the energy deficit which must be supplied by the thermal energy stored in the reservoir. In this case, the power plant is sized to meet exactly the base's total energy demands, and the unshaded area between the thermal energy supply and demand schedules (the amount of excess thermal energy available) is equal to the shaded area, indicating that there is no net drain or storage of energy in the reservoir during the 24-hour period. The reservoir, shown by the dotted curve, experiences a single complete duty cycle involving approximately 72,000 cubic feet of its capacity, and no auxiliary cooling is required since no energy is produced in excess of that required by the consumers. These results indicate clearly that the 150 MWt plant can supply the base's peak winter energy demands as predicted by the preliminary simulation analyses.

The peak summer day simulation results are shown in Fig. 5.35. In this case, the central station thermal power

Figure 5.35

75% Thermal/Electrical Load Split
Utility System Design Verification
Peak Summer Day Simulation



output has been reduced to the 140 MWt necessary to meet the peak electrical demand, and the system is over-supplied with thermal energy. The shaded area between the thermal energy supply and demand schedules is the deficit which must be supplied by stored heat, dictating the installation of a reservoir with a volume of roughly 133,000 cubic feet, having hot and cold water section temperatures of 380 °F and 151 °F, respectively. The unshaded area between the thermal energy supply and demand schedules represents the excess thermal energy produced by the power plant over that consumed directly by the utility system. After re-charging the reservoir, the net surplus is approximately 635 MWhr. The reservoir duty cycle is shown by the dotted curve, the flat portion of which during the period from 1 AM to 8 AM indicates that the reservoir is completely charged and that the surplus thermal energy produced during these hours must be dissipated in the auxiliary cooling system. The discrepancy of approximately 9,000 cubic feet in the reservoir size between the results of this simulation and the volume quoted in Section 5.1.6.3 is within the accuracy of the rating calculations, and the proposed system component sizing data is confirmed.

REFERENCES

1. Personal communication, Capt. Anthony von Nida, USAFESA, Fort Belvoir, Va.; forwarded from Maj. R.E. Dettling, Acting Chief, USAF Global Environmental Applications Branch, (1976).

2. Personal communication, Capt. Anthony von Nida, USAFESA, Fort Belvoir, VA., (1976).

3. Kusuda, T., "SUN, An Algorithm to Find Solar Position, and Intensity of Direct Normal and Diffuse Radiation," NBSLD, Computer Program for Heating and Cooling Loads in Buildings, NBSIR-74-574, 6a-14a, National Bureau of Standards, Washington, D.C., (Nov. 1974).

4. Feher, R.D., "Energy Storage Alternatives for a Nuclear Total Utility System," MIT Department of Nuclear Engineering Special Study Report, (May 1974.)

5. "An Assessment of Energy Storage Systems Suitable for Use by Electric Utilities," Volume 2, Public Service Electric and Gas Company, Newark, EPRI Research Project 225, (Dec. 1975.)

CHAPTER 6

ECONOMIC OPTIMIZATION

6.1 Introduction

The economic optimization of the TES is discussed in four parts, the economics of the HTGR as developed by Metcalfe, et al. [1], the economics of the CGGT plant, the economics of the TUS, and the economics of the combined Total Energy System. The standard of economic comparison used in this report is the cost of the proposed system in 1985 dollars. However, the data base and escalation rates used to project the 1985 costs are also presented.

6.2 HTGR Costs

The size of the HTGR required to power the Ft. Bragg TES is determined by the winter peak design day. For Ft. Bragg the maximum load on the TES occurs on the design winter day, the day with the lowest temperature profile of the year. More southernly locations would be expected to display reductions in the size of the winter peak and increases in the size of the summer peak load requirements. Reactor capacity would then be determined by the summer peak design day temperature.

In addition to the exterior temperature schedule, the thermal/electric load split also has an important effect on the required reactor size. Consider the case of a reactor supplying power to a 100% thermal TUS (such a TUS is sized to meet 100% of the thermal space-conditioning demand on the base throughout the year). The reactor would be generating electrical power for the non-space-conditioning electrical demand (i.e. lights, refrigerators, fans, etc.) and thermal power for space heating or cooling. For Ft. Bragg's climate on the winter design day, the "waste heat" (produced as a result of generating the non-space conditioning electrical demand) is less than the thermal energy required to heat the structures on the base. The reactor capacity must therefore be made larger than that needed for generation of the electrical power demanded by the non-space conditioning electrical load, in order to supply the required thermal demand.

Now consider the case of a reactor powering a TUS in which some of the home heating is performed by heat pumps as described in Chapters 4 and 5. Because some of the consumers are using heat pumps to heat their homes, as the number of heat pump-using homes increases (increasing the value of the electrical split), the amount of heat drawn from the atmosphere increases and the amount of direct heating that must be supplied by the reactor decreases. That is, although the total heating demand remains constant, as the number of heat-pumps increases the amount of heat drawn from the atmosphere increases and the net amount supplied by the reactor decreases. Thus, the needed reactor capacity decreases as the percent of heat pumps increases. It is seen that the reactor size continues to decrease as the electric heat pump split of the TUS increases until the waste heat produced from the generation of electrical power matches the thermal demand. Beyond this unique demand split value, the waste heat generated will be larger than the thermal demand and reactor size again will begin to increase to meet the required electrical demand. Figure 6.1 illustrates the variation of the required reactor size (computed by TDIST) versus TUS thermal demand share. As the value of the thermal/electric load split is reduced the required reactor size is seen to decrease, also. The minimum reactor size occurs at a thermal/electric load split value of 75%. At this value of thermal/electric split the reactor power plant waste heat and electrical output exactly match the thermal and electrical demands, and no energy is wasted.

As the thermal/electric load split value is reduced further, the required increase in electrical output (and hence reactor size) results in increased reactor waste heat output which exceeds the (now) reduced TUS demand and is, therefore, wasted energy.

Figure 6.2 shows the capital cost of the HTGR in 1985 dollars computed from Metcalfe's work as detailed in Appendix D.1. Also shown are the fuel cycle cost, operation and maintenance costs, and the cost of a fossil-fired auxiliary boiler

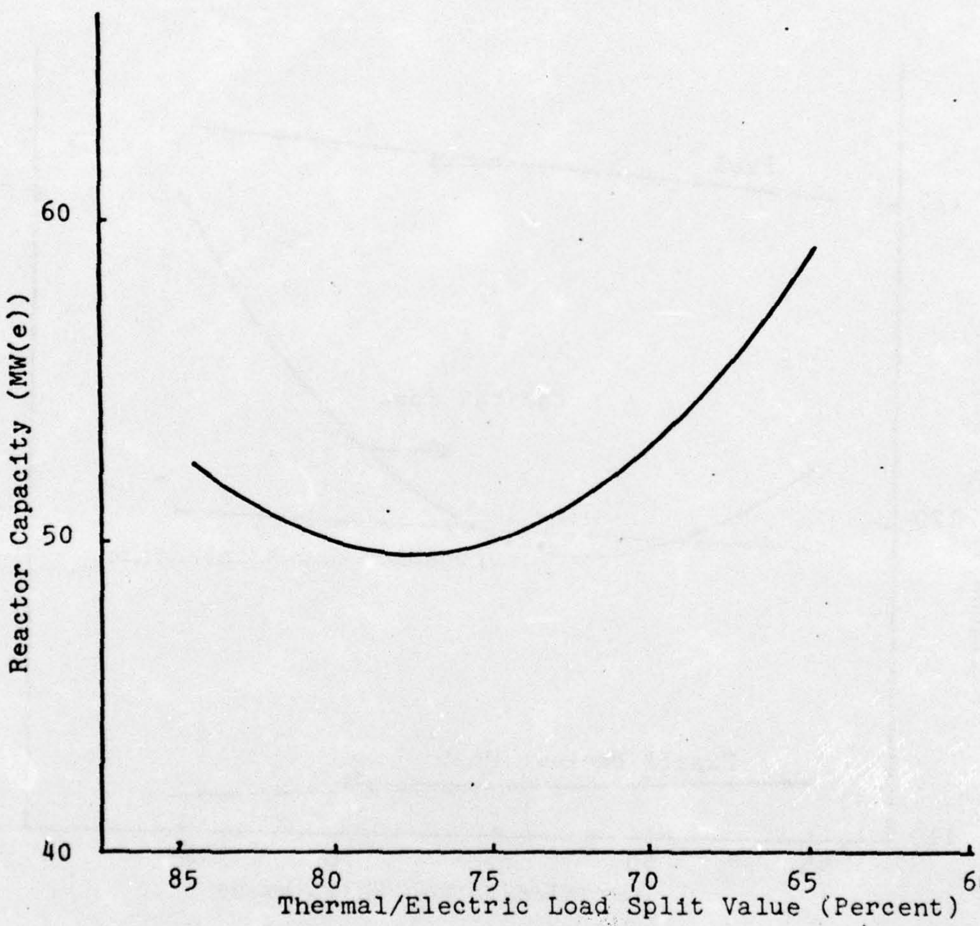


Figure 6.1. Required Reactor Capacity vs. Thermal/Electric Load Split Value

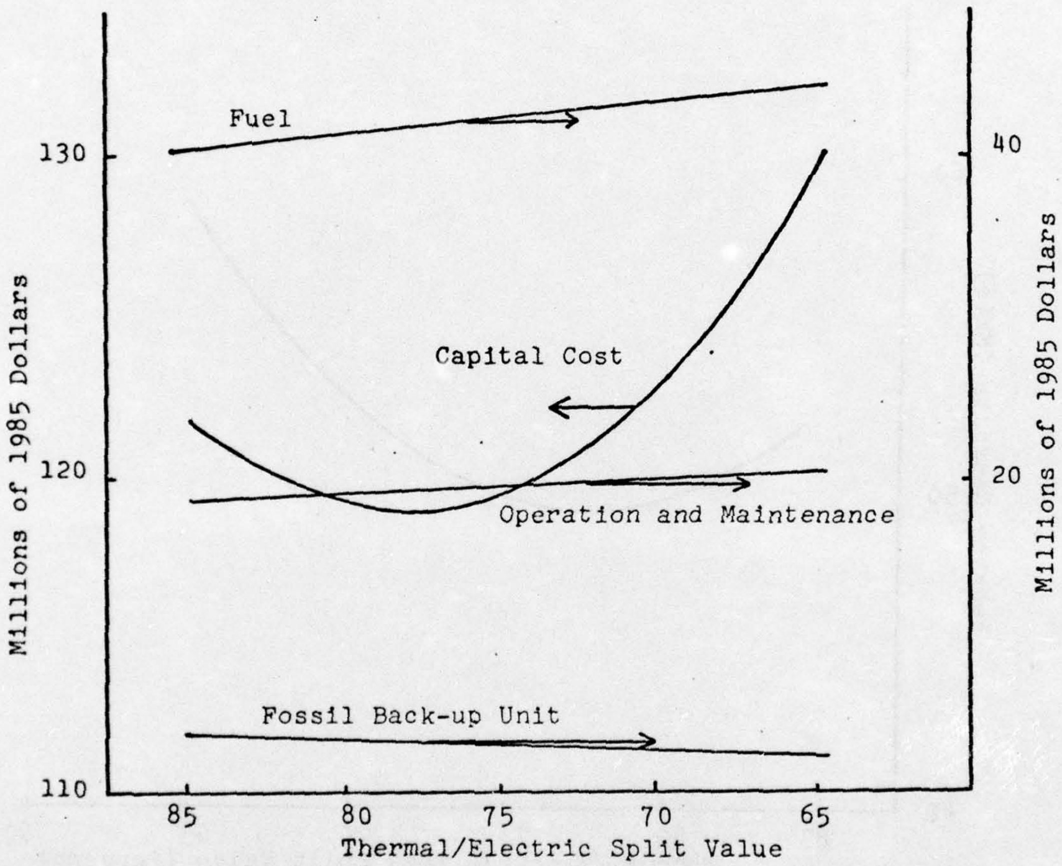


Figure 6.2. HTGR Power Station Present Worth Costs vs. Thermal/Electric Load Split Value

to supply back-up power to the TUS.

Detailed calculations of HTGR capital cost are discussed in Appendix D.1 but, briefly, the capital cost is calculated by evaluating the functional dependence of capital cost versus size as given by Metcalfe, using the CONCEPT III code [1]. It is seen that the HTGR capital costs vs. thermal/ electrical load split data display less curvature than the HTGR size vs. thermal/electric split data. This broadening is due to the economy-of-scale that results in larger HTGRs being less expensive on a unit capacity cost basis than smaller HTGRs. Thus, the optimal (i.e. smallest) HTGR is the least expensive in terms of total cost, but the most expensive in terms of unit capacity cost.

The cost of the fossil-fired back-up power plant for the TUS decreases with decreasing values of thermal/electric load split, due to the reduced size of the TUS. Similarly, as the load split value decreases, increased fuel and operation and maintenance costs are observed, which reflect the increase in required electrical energy output.

6.3 Coal Gasification Gas Turbine Costs

Table 6.1 lists the important parameters of the CGGT components. Estimating the cost of gasification equipment is complicated by the dependence of equipment cost on coal type and by the reluctance of vendors to commit themselves to unit

TABLE 6.1

MAJOR POWER PLANT COMPONENT COSTS AND PARAMETERS

Lurgi Gasifier Units Specifications

Daily Gas Production	8.00×10^9 BTU/Day
Gas Heating Value	122 BTU/SCF
Coal Consumption Rate	526 Tons/Day
Coal Heating Value	9,500 BTU/lb
Water Consumption Rate	1.11×10^5 gal/Day
Estimated Capital Costs for One Gasifier Unit[1] 1975	\$7.5 million
Assumed Capital Cost Inflation Rate	1970-1973, 8%
	1974-1975, 22%
	1975-1985, 6.3%[2]

Turbo Power Marine FT4C Power Plant, with HTW Waste Heat Exchanger Specifications

Electrical Capacity	26.3 Mw(e)
Waste Heat Recovery Rate	32.0 Mw(t)
Typical Electrical Generation Heat Rate	13.5×10^3 BTU/KW-hr
Estimated Capital Costs for One FT4C Unit:	
1975	\$4.4 million
1985	\$8.1 million
Assumed Capital Cost Inflation Rate	1975-1985, 6.3%

[1] Cost estimate based upon data presented in "Clean Fuel Gas from Coal," Lurgi Corporation, publication number 0 1007/ 10,71, 1970.

[2] Escalation rates for 1970-1975 as recommended by gasifier vendors [3]; escalation rates for 1975-1985 in conformity with rates used by Metcalfe [1].

cost data. Projecting costs into the future is further complicated by the uncertainty in escalation rates. Escalation rates projected by Metcalfe are used to facilitate comparisons between different power plant types and to insure uniformity between the two reports. The cost of a CGGT system for a given TUS split is determined by matching the capacities of the components to the thermal/electric load calculated by TDIST. Because the gasifiers and gas turbines are only available in certain sizes, the capital cost of the CGGT plant does not vary continuously with TUS split, but instead changes incrementally as each additional module is added into the system.

Coal consumption is calculated by using the following technique: (detailed in Appendix D.2)

1) A twenty-four hour simulation of Ft. Bragg thermal and electrical power demands for a particular day at a given thermal/electric load split is performed (Figure 5.34),
2) The gas consumption required to generate the electrical demand schedule is calculated by using an average heat rate for the gas turbine generators,
3) The waste heat recovered from the turbine exhaust is subtracted from the total thermal energy demand calculated by TDIST - if the total thermal energy demand exceeds the waste heat recovered from the gas turbine additional gas is burned in a central hot water heater,

4) The total amount of coal consumed for the day is found by adding the electrical gas consumption to any extra heating gas consumption, and converting gas consumption to coal consumption via the gasifier conversion efficiency,

5) The yearly coal consumption for a given thermal/electric load split is found by repeating steps 1 through 4 over the desired range of annual weather variation. This provides the basic data for the annual fuel consumption integration. In practice, an average winter day, an average winter-spring day, an average spring-summer day and an average summer day simulations are used to construct an annual fuel consumption curve (see Figure 6.3). The annual fuel consumption curve is then integrated over the year to obtain total annual fuel consumption. Steps 1 through 5 must be repeated for each thermal/electric split of interest. Additionally, the winter peak and summer peak design day simulations must be performed, since these days determine the TES maximum load and hence the required equipment capacities.

Figure 6.3 shows the annual fuel consumption for three different values of thermal/electrical split. These curves show the expected decrease in fuel consumption from winter

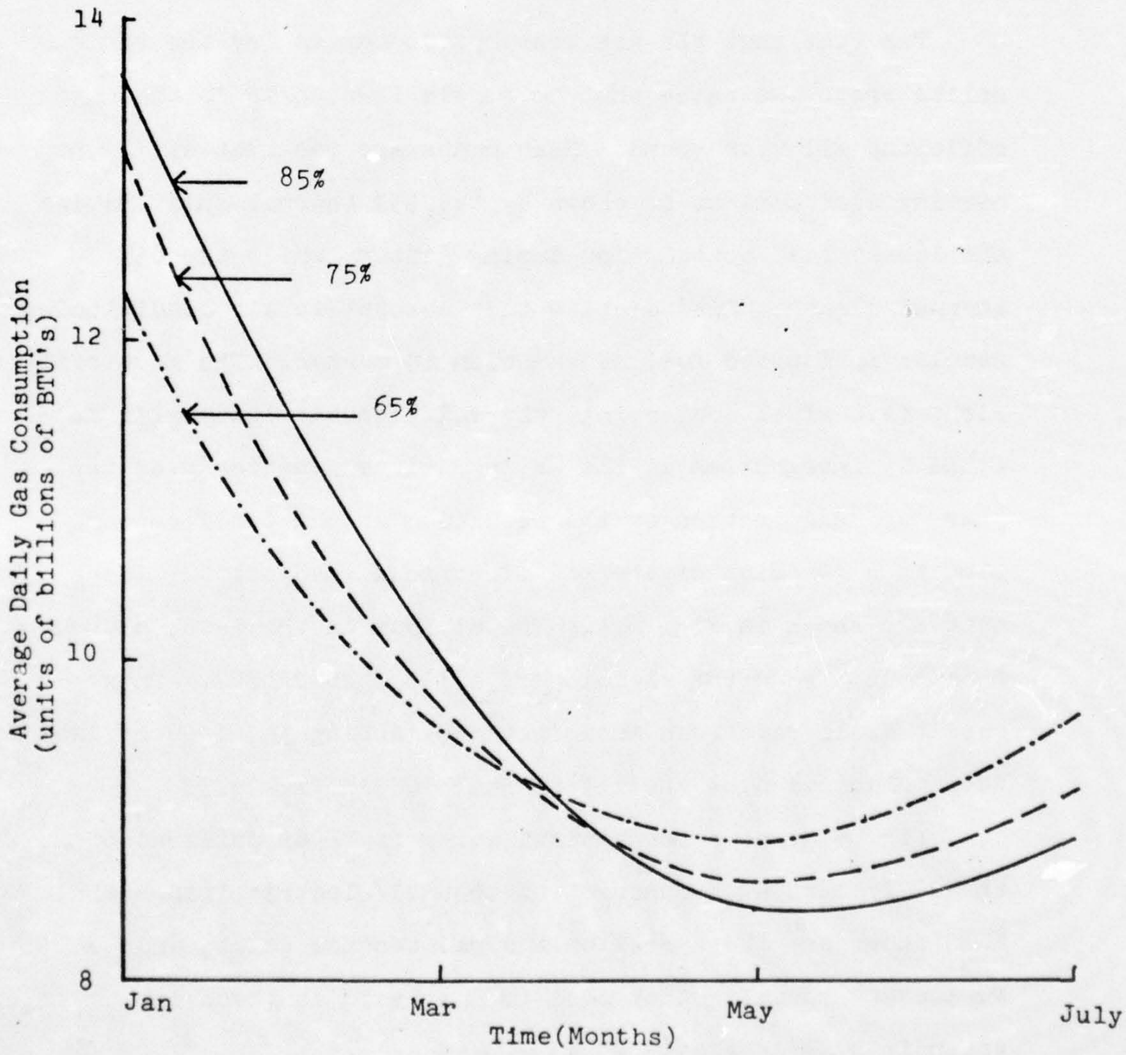


Figure 6.3. Annual Variation of Fuel Consumption As a Function of Thermal/Electric Load Split

AD-A031 744 MASSACHUSETTS INST OF TECH CAMBRIDGE DEPT OF NUCLEAR--ETC F/G 13/1
DESIGN OF A NUCLEAR POWERED TOTAL ENERGY SYSTEM FOR FORT BRAGG,--ETC(U)
MAY 76 J W STETKAR, F R BEST, M W GOLAY DAAK02-74-C-0308

UNCLASSIFIED USAFESA-RT-2011 NL

3 OF 4
AD
A031744

to spring and then the increase from spring to summer. The winter maximum in gas consumption is due to the heating demand on the TES, and the summer maximum in gas consumption is due to the air conditioning load.

The fact that the gas consumption curves for the various splits cross indicates that no single load split is the most efficient all year round. Heat pumps are the most efficient heating equipment as is shown by the 65% thermal split having the lowest fuel consumption during winter, while the 85% thermal/electric load dictate that absorptive air conditioning results in lowered fuel consumption in summer. The most efficient (i.e. fuel-conserving) thermal/electric load split is found by integration of the daily fuel consumption over the year, and examination of the resulting annual total consumption as a function of thermal/electrical load split. These data are shown in Fig. 6.4. The minimum in the data, although broad, occurs in the vicinity of a 75% thermal TUS. This result could have been anticipated by noting the intermediate fuel consumption of the 75% thermal TUS in Fig. 6.3.

Fig. 6.5 shows the capital costs (in 1985 dollars) of the CGGT plant as a function of thermal/electric load split. Also shown are the operation and maintenance costs, and fuel costs assuming that coal (of analysis as given in Appendix D.3) is available at $27/ton.

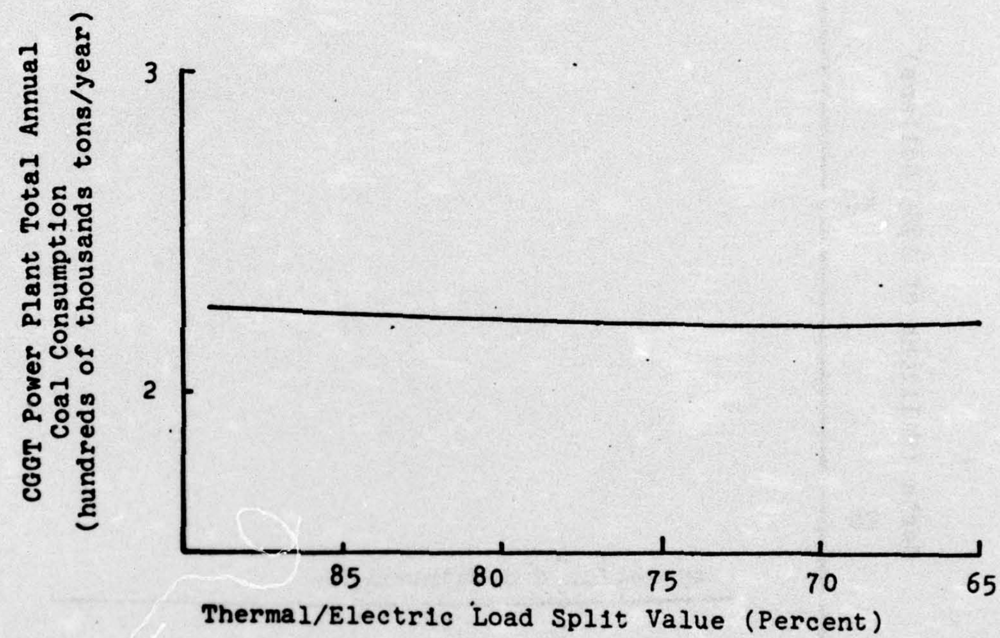
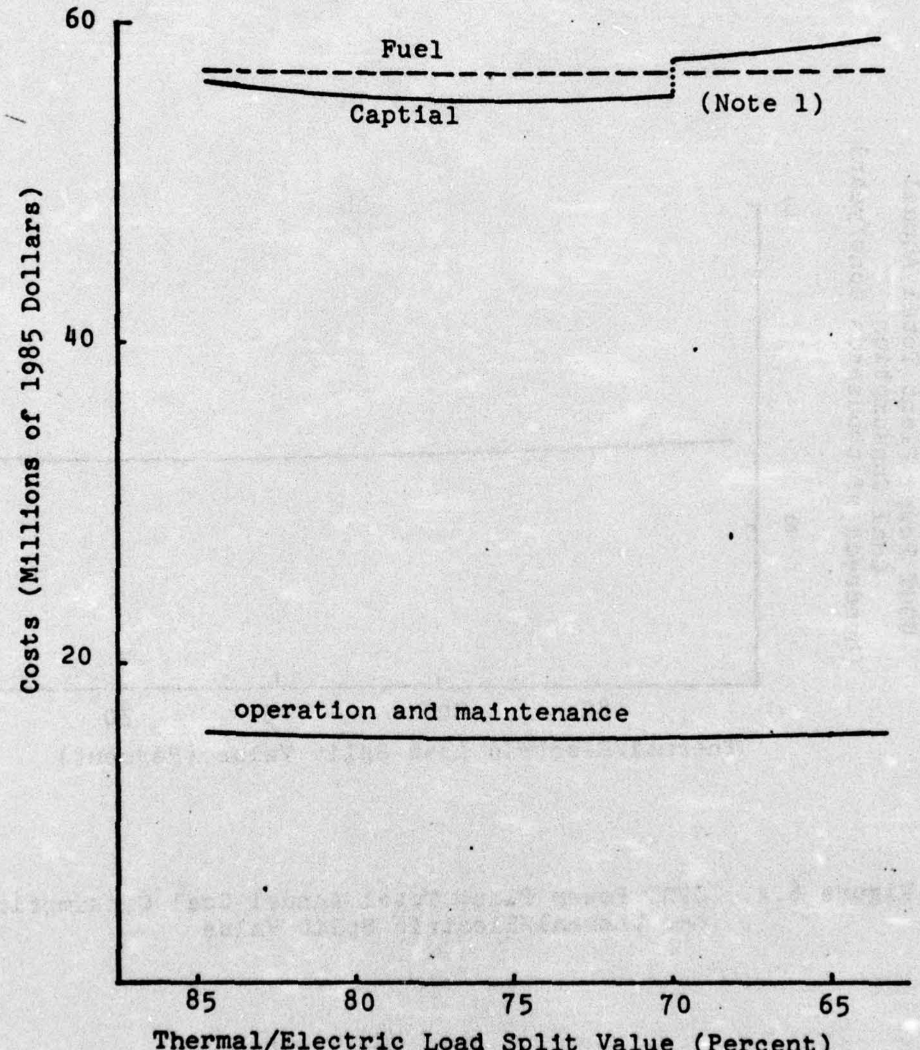


Figure 6.4. CGGT Power Plant Total Annual Coal Consumption vs. Thermal/Electric Split Value

Figure 6.5. Power Station Present Worth Costs vs. Thermal/Electric Load Split Value



1. The discontinuity in Capital Cost is due to incremental addition of one FT4C turbine.

6.4 Thermal Utility System Costs

6.4.1 Pumps and Piping

The three primary components of the thermal utility system piping costs are: (1) the capital costs of the pipe and its associated trenching, (2) the pumping power costs over the 30-year life of the system, and (3) the capital costs of the pumps. Figure 6.6 illustrates the variation of the installed pipe cost with pipe diameter. The base cost data used to produce this curve are presented in Appendix D.6. Since it is assumed that all primary and secondary piping in the system consists of an insulated supply line and an uninsulated return line buried in a common trench, the cost of one mile of piping as shown in the figure includes the cost of one mile each of insulated and uninsulated pipe and the cost for excavation and backfilling of a one mile trench to contain both pipes at a burial depth of 6 feet on center.

For a given fluid mass flowrate, as the pipe size is increased, the fluid velocity and the associated frictional pressure losses decrease, reducing the required rating of the circulation pump and decreasing the total pumping power needed to drive the fluid around the loop. Figure 6.7 illustrates the variation of pumping power costs as a function of pipe size for several representative fluid mass flowrates. In estimating these costs, average pump efficiencies of 70%

Figure 6.6

Installed Piping System Cost vs. Pipe Size
(Includes one insulated and one uninsulated pipe buried in a common trench)

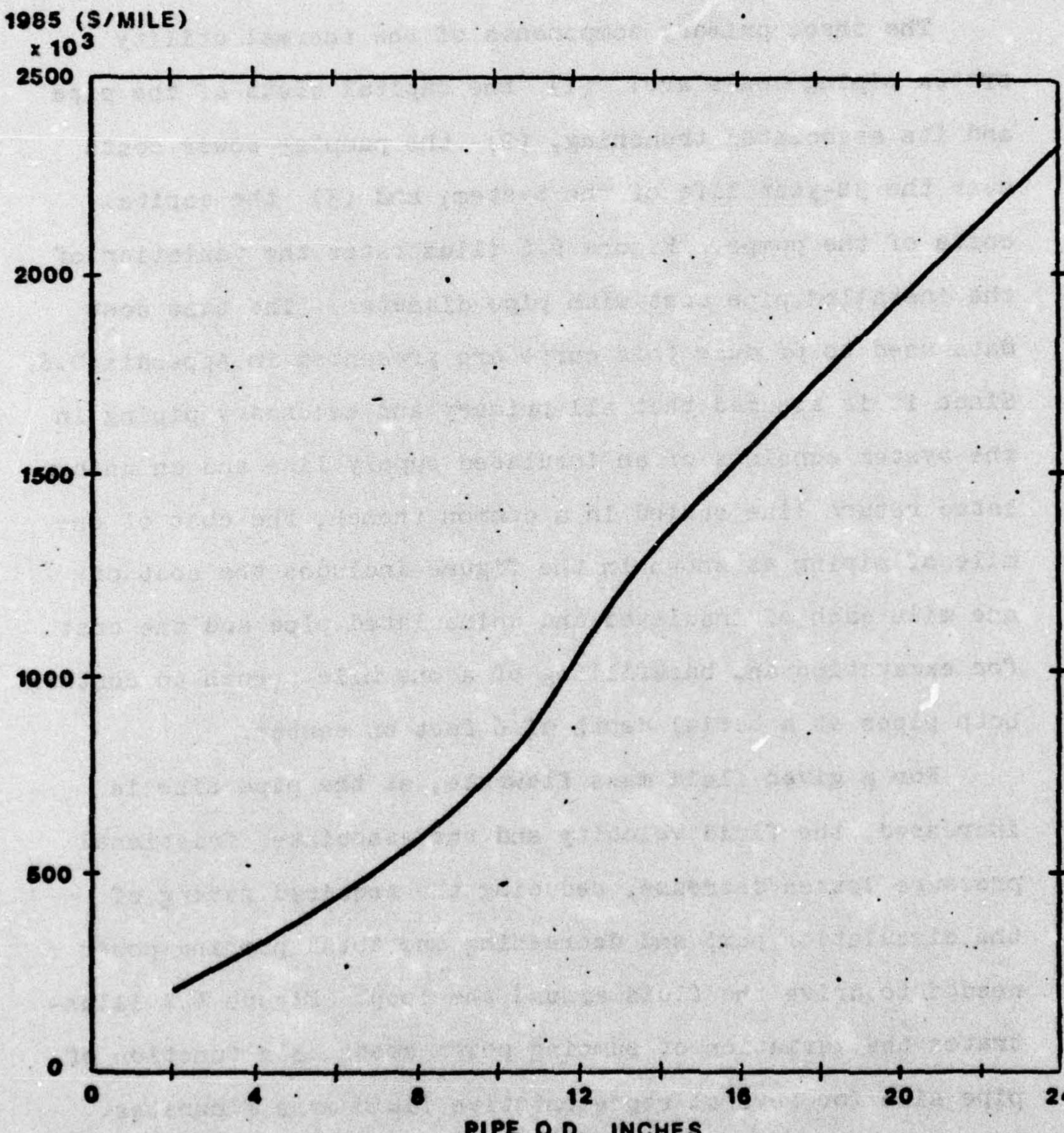
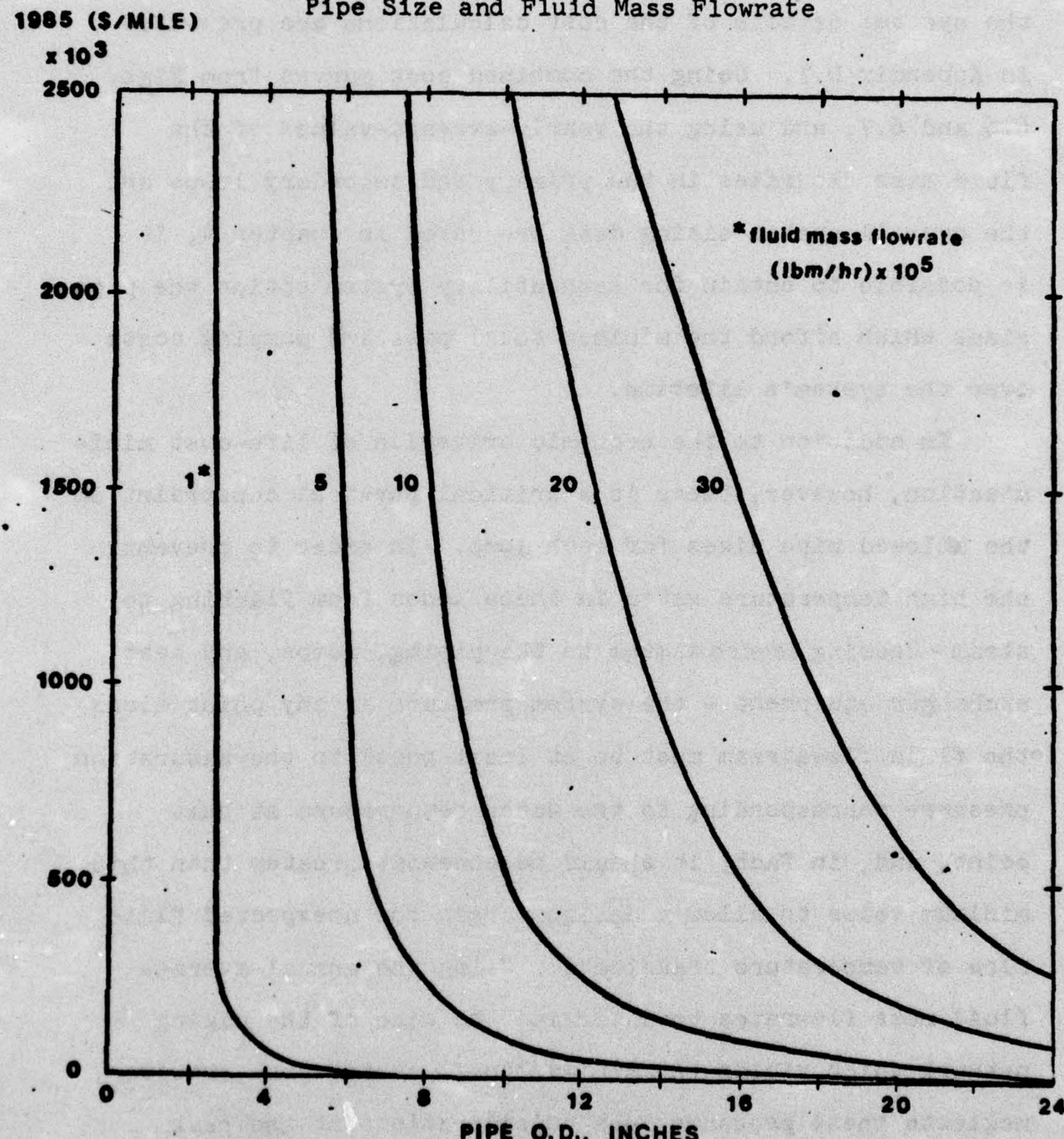


Figure 6.7

Pumping Power Cost Over Life
as a Function of
Pipe Size and Fluid Mass Flowrate



are assumed and a levelized cost of electricity of 58 mills/KWhr is assumed to obtain throughout the 30-year life of the system; details of the cost calculations are presented in Appendix D.7. Using the combined cost curves from Figs. 6.6 and 6.7, and using the yearly-average-values of the fluid mass flowrates in the primary and secondary loops and the overall system sizing data presented in Chapter 4, it is possible to obtain for each utility system option the pipe sizes which afford the minimum total pipe and pumping costs over the system's lifetime.

In addition to the economic criterion of life-cost minimization, however, there is a critical physical constraint on the allowed pipe sizes for each loop. In order to prevent the high temperature water in these loops from flashing to steam - causing severe damage to the piping, pumps, and heat exchanger equipment - the system pressure at any point along the fluid flowstream must be at least equal to the saturation pressure corresponding to the water temperature at that point, and, in fact, it should be somewhat greater than this minimum value to allow a design margin for unexpected fluid flow or temperature transients. Using the annual-average fluid mass flowrates to determine the size of the piping network which yields the minimum total system cost, however, neglects these pressure loss considerations at the peak system flows, and it has been found that the pipes selected according to the cost minimization criterion are uniformly unacceptable for use, due to the excessive pressure losses exhibited during periods of high thermal loading. Therefore, in selecting the final piping parameters for each of the three system options, the primary design criterion followed has been that - under the design peak thermal energy demand conditions - the fluid frictional pressure loss around any loop should not exceed approximately 30% of the maximum saturation pressure to be maintained in that loop. (e.g., If the primary loop water temperature varies from 380 °F at the inlet to 150 °F at the outlet, with a 50 psia design margin added to the minimum saturation pressures, the system pressure must be at least 250 psia at the inlet and 50 psia at the outlet. The maximum frictional pressure loss allowed around this loop is 30% of 250 psia, or approximately 80 psia.) Tables 6.2-6.4 present the final pipe design criteria and costs for each of the utility system options studied. It should be noted that in some instances the choice of an intermediate pipe size somewhat smaller than that listed would reduce the design pressure margin and would similarly reduce the associated pipe costs for the affected loop. However, due to cost data limitations and a lack of technical specifications for any other pipe sizes, only the standard pipes shown in the tables have been considered for installa-

TABLE 6.2

85% THERMAL/ELECTRICAL LOAD SPLIT OPTION

THERMAL UTILITY SYSTEM PIPING COSTS

Subsystem	Loop(1)	Maximum Design Mass Flowrate (lbm/hrx10^5)	Maximum Saturation Pressure (psia)	Maximum Pressure Loss (psia)	Pipe O.D. (inches)	Length(2) (miles)	In- Place Pipe Cost (1985 \$x10^6)
Primary	Primary	30.31	250	40.1	24	10.0	11.46
Secondary	1	7.56	200	45.4	12	6.4	3.45
	2	8.53	100	13.0	18	7.8	6.72
	4	14.48	75	6.6	24	6.2	7.11
Tertiary		(3)	70-75	(3)	8	5.3	1.53
		(3)	70-75	(3)	6	32.7	7.62
		(3)	70-75	(3)	4	34.4	5.50
		(3)	70-75	(3)	3	5.6	0.83
		(3)	70-75	(3)	2	80.4	8.83
Total							53.05

(1) See Fig. 4.3.

(2) Total length of loop includes insulated supply line and uninsulated return line.

(3) Maximum fluid velocity not to exceed 10 ft/sec in any branch.

TABLE 6.3

75% THERMAL/ELECTRICAL LOAD SPLIT OPTION

THERMAL UTILITY SYSTEM PIPING COSTS

Subsystem	Loop[(1)]	Maximum Design Mass Flowrate ($lbm/hr \times 10^5$)	Maximum Saturation Pressure (psia)	Maximum Pressure Loss (psia)	Pipe O.D. (inches)	Length[(2)] (miles)	In - Place Pipe Cost (1985 $\$ \times 10^6$)
Primary	Primary	24.72	250	27.8	24	10.0	11.46
Secondary	1	5.38	200	63.7	8	2.4	0.69
	2	8.24	120	12.3	18	7.8	6.72
	4	15.19	75	7.2	24	6.2	7.11
Tertiary		(3)	70-75	(3)	8	5.3	1.53
		(3)	70-75	(3)	6	21.0	4.89
		(3)	70-75	(3)	4	31.0	4.95
		(3)	70-75	(3)	3	5.6	0.83
		(3)	70-75	(3)	2	65.5	7.20
Total							45.38

(1) See Fig. 4.4.

(2) Total Length of Loop includes insulated supply line and uninsulated return line.

(3) Maximum fluid velocity not to exceed 10 ft/sec in any branch.

TABLE 6.4

65% THERMAL/ELECTRICAL LOAD SPLIT OPTION

THERMAL UTILITY SYSTEM PIPING COSTS

Subsystem	Loop(1)	Maximum Design Mass Flowrate ($lbm/hr \times 10^5$)	Maximum Saturation Pressure (psia)	Maximum Pressure Loss (psia)	Pipe O.D. (inches)	Length(2) (miles)	In-Place Pipe Cost (1985 $\$ \times 10^6$)
Primary	Primary	22.04	250	22.6	24	10.0	11.46
Secondary	2	8.62	200	16.5	18	9.7	8.36
	4	13.23	80	22.8	18	6.2	5.34
Tertiary		(3)	70-75	(3)	8	5.3	1.53
		(3)	70-75	(3)	6	16.3	3.80
		(3)	70-75	(3)	4	26.9	4.30
		(3)	70-75	(3)	3	5.6	0.83
		(3)	70-75	(3)	2	58.4	6.42
Total							42.04

(1) See Fig. 4.5.

(2) Total length of loop includes insulated supply time and uninsulated return line.

(3) Maximum fluid velocity not to exceed 10 ft/sec in any branch.

tion. The tertiary piping, which has not been studied in any of the TDIST simulations, is sized such that the maximum fluid velocity in any branch does not exceed 10 feet per second under the peak thermal loading conditions.

Using the peak fluid mass flowrates and the piping sizes listed in Tables 6.2-6.4 for each of the system options, the ratings and costs of the primary and secondary loop circulation pumps can be determined. (See Appendix D.7 for details of the rating and cost data). In order to compute the total energy consumed for the operation of these pumps, the annual-average fluid flowrates for each of the piping loops are assumed to obtain throughout the system's operating lifetime of 30 years. Appendix D.7 contains details of the pumping power cost calculations, the results of which are summarized in Tables 6.5-6.7. The pump ratings are determined by the design peak fluid flowrates in each of the piping loops. However, because the annual-average energy demands for the systems are much less than the summer and winter design peaks, the pumps operate at low capacity factors over much of the year, and the total present-worth costs of the energy consumed by them over the system lifetime are small compared with other system cost components.

6.4.2 Heat Exchangers

The heat exchanger models used in the TDIST simulations are described in Chapter 4 as being single-pass, counterflow

TABLE 6.5

85% THERMAL/ELECTRICAL LOAD SPLIT OPTION

PUMP AND PUMPING POWER COSTS

Subsystem	Loop[1]	Annual Average Mass Flowrate ($1bm/hr \times 10^5$)	Pumping Power (KW)	Lifetime Pumping Power Cost[2] (1985 $\$ \times 10^3$)	Pump Rating (gpm)	Pump Cost (1985 $\$ \times 10^3$)	Total Cost[3] (1985 $\$ \times 10^3$)
Primary	Primary	5.65	.996	21.7	6300[6]	37.0[6]	58.7
Secondary	1	1.28	.215	4.7	1600	10.3	15.0
	2	1.45	.070	1.5	1800	11.7	13.2
	4	2.32	.052	1.1	3000	16.9	18.0
Tertiary[4]							
Total				29.0[5]		75.9[5]	104.9[5]

(1) See Fig. 4.3.

(2) Total cost over a 30 year lifetime assuming 70% efficient pumps, a levelized electricity cost of 58 mills/KWhr, and continuous operation at the average flow conditions.

(3) Total of pumping power and pump capital costs.

(4) Tertiary piping not included due to a lack of detailed annual flow data.

(5) For conservatism, total costs shown may be doubled to include tertiary system components.

(6) Use of two 3150 gpm pumps assumed due to lack of cost data for units sized above 3500 gpm.

TABLE 6.6

75% THERMAL/ELECTRIC LOAD SPLIT OPTION

PUMP AND PUMPING POWER COSTS

Subsystem	Loop(1)	Annual Average Mass Flowrate ($1bm/hr \times 10^5$)	Pumping Power (KW)	Lifetime Pumping Power Cost(2) (1985 $\$ \times 10^3$)	Pump Rating (gpm)	Pump Cost (1985 $\$ \times 10^3$)	Total Cost(3) (1985 $\$ \times 10^3$)
Primary	Primary	4.88	.661	14.4	5100(6)	30.9(6)	45.3
Secondary	1	0.88	.195	4.2	1100	8.6	12.8
	2	1.38	.061	1.3	1700	11.0	12.3
	4	2.31	.050	1.1	3200	18.5	19.6
Tertiary(4)							
Total				21.0(5)		69.0(5)	90.0(5)

(1)See Fig. 4.4.

(2)Total cost over a 30 year lifetime assuming 70% efficient pumps, a levelized electricity cost of 58 mills/KWhr, and continuous operation of the average flow conditions.

(3)Total of pumping power and pump capital costs.

(4)Tertiary piping not included due to a lack of detailed annual flow data.

(5)For conservatism, total costs shown may be doubled to include tertiary system components.

(6)Use of two 2550 gpm pumps assumed due to lack of cost data for units sized above 3500 gpm.

TABLE 6.7

65% THERMAL/ELECTRICAL LOAD SPLIT OPTION

PUMP AND PUMPING POWER COSTS

Subsystem	Loop[1]	Annual Average Mass Flowrate (lbm/hrx10^5)	Pumping Power (KW)	Lifetime Pumping Power Cost[2] (1985 \$x10^3)	Pump Rating (gpm)	Pump Cost (1985 \$x10^3)	Total Cost[3] (1985 \$x10^3)
Primary	Primary	3.55	.271	5.9	4600[6]	28.8[6]	34.7
Secondary	2	1.48	.093	2.0	1800	11.7	13.7
	4	2.36	.219	4.7	2700	15.5	20.2
Tertiary[4]							
Total				12.6[5]		56.0[5]	68.6[5]

(1) See Fig. 4.5.

(2) Total cost over a 30 year lifetime assuming 70% efficient pumps, a levelized electricity cost of 58 mills/KWhr, and continuous operation at the average flow conditions.

(3) Total of pumping power and pump capital costs.

(4) Tertiary piping not included due to a lack of detailed annual flow date.

(5) For conservatism, total costs shown may be doubled to include tertiary system components.

(6) Use of two 2300 gpm pumps assumed due to lack of cost data for units sized above 3500 gpm.

units with 1" O.D. tubes and 1" I.D. channels in square-bundle matrices, 80 feet long. Due to a lack of detailed design and cost data for commercially-available heat exchangers of the ratings required for the utility system primary and secondary loops, no attempt has been made to translate the model design parameters into actual component sizing and performance characteristics. The cost function used in computing the capital costs of these heat exchangers - given in Eq. (6.1) - is relatively insensitive to the details of the heat exchanger configuration, requiring only a specification of the total heat transfer area of the unit.

$$C_x = 506{,}000 + 5.9A \qquad (6.1)$$

where C_x = heat exchanger cost (1985 \$), and
A = total heat transfer area (ft^2).

Although originally derived for 1971 costs [4], the coefficients in Eq.(6.1) have been escalated to 1976 dollars through the use of the Nelson cost index of 1.23 [4] and have been scaled further from 1976 to 1985 dollars using the 6.2% annual escalation rate recommended by Metcalfe [1]. Tables 6.8-6.10 present the heat exchanger costs for each of the three utility system options. No estimates have been made of the number, sizes, or costs of any heat

TABLE 6.8

85% THERMAL/ELECTRICAL LOAD SPLIT OPTION

HEAT EXCHANGER COSTS

Loop(1)	Heat Exchanger(1)	Heat Transfer Coefficient (BTU/hr°Fx10^6)	Heat Transfer Area (ft^{2}x10^3)	1985 Cost (\$x10^5)
Primary	1	3.74	24.0	6.48
	2	3.86	24.9	6.53
	4	10.85	69.7	9.18
1	5	1.33	8.5	5.57
	6	1.33	8.5	5.57
	7	0.79	5.1	5.36
	8	0.79	5.1	5.36
	13	1.33	8.5	5.57
2	9	1.09	7.0	5.47
	10	2.25	14.4	5.91
	11	2.25	14.4	5.91
	12	2.25	14.4	5.91
4	18	2.03	13.0	5.83
	19	6.55	42.0	7.54
	20	6.55	42.0	7.54
	21	2.03	13.0	5.83
	22	6.55	42.0	7.54
Total				107.10

(1) See Fig. 4.3.

TABLE 6.9

75% THERMAL/ELECTRICAL LOAD SPLIT OPTION

HEAT EXCHANGER COSTS

Loop[1]	Heat Exchanger[1]	Heat Transfer Coefficient (BTU/hr°Fx10^6)	Heat Transfer Area (ft^{2}x10^3)	1985 Cost (x10^5$)
Primary	1	2.49	16.0	6.01
	2	3.93	25.2	6.55
	4	12.95	83.2	9.97
1	5	1.28	8.2	5.55
	6	1.28	8.2	5.55
	13	1.28	8.2	5.55
2	9	0.92	5.9	5.41
	10	1.90	12.2	5.78
	11	1.90	12.2	5.78
	12	1.90	12.2	5.78
4	18	2.03	13.0	5.83
	19	6.30	40.4	7.45
	20	6.30	40.4	7.45
	21	2.03	13.0	5.83
	22	6.30	40.4	7.45
Total				95.94

(1) See Fig. 4.4.

TABLE 6.10

65% THERMAL/ELECTRICAL LOAD SPLIT OPTION

HEAT EXCHANGER COSTS

Loop[1]	Heat Exchanger[1]	Heat Transfer Coefficient (BTU/hr°Fx10^6)	Heat Transfer Area (ft^2x10^3)	1985 Cost (x10^5$)
Primary	2	4.72	30.3	6.85
	4	9.33	59.9	8.60
2	5	1.33	8.5	5.57
	9	0.62	4.0	5.30
	10	1.33	8.5	5.57
	11	1.33	8.5	5.57
	12	1.33	8.5	5.57
4	18	1.61	10.3	5.67
	19	5.19	33.3	7.03
	20	5.19	33.3	7.03
	21	1.61	19.3	5.67
	22	5.19	33.3	7.03
Total				75.46

(1)See Fig. 4.5.

exchangers within individual buildings.

6.4.3 Thermal Energy Storage Reservoir

In computing the costs of the thermal energy storage reservoir for each system option, the 1973 benchmark cost of $1 per gallon of storage capacity used by Nida [5] is doubled for conservative estimation and is escalated at 6.2% per year to $4.12 per gallon in 1985. Since the reservoir is composed of an interconnected set of from 2 to 11 identical tanks, it is assumed that this unit cost applies uniformly to all designs, independently of their total storage volume. Table 6.11 lists the costs of the three units considered for installation.

6.4.4 Auxiliary Cooling Tower Costs

Under normal system operating conditions, the maximum rate of thermal energy dissipation in the auxiliary cooling system is determined by the peak mismatch between the thermal energy supply and the demand occurring on a late spring or average summer day. (See Section 5.5 for a more detailed discussion of the cooling system ratings.) However, for conservatism in computing the costs of these units, it is assumed that the cooling system will be designed to dissipate the maximum thermal power output of the TES central station power plant with which it is associated. Since auxiliary cooling has not been studied

TABLE 6.11

THERMAL ENERGY STORAGE RESERVOIR COSTS

Thermal/ Electrical Load Split Option	Reservoir Volume (galx10^3)	Number of Tanks	1985 Cost(1) (\$x10^6)
85%	1809	11	7.45
75%	987	6	4.06
65%	329	2	1.35

(1) Base cost of \$2/gal in 1973, escalated to \$4.12/gal in 1985.

in any of the TDIST simulations, no recommendations are made as to the type of system to be installed at the Fort Bragg site, but it is assumed for cost estimation purposes that a set of modular mechanical draft cooling towers will be employed to provide the necessary capacity. Present-day costs of such towers lie in the range of $30 million for a series of units to serve a 1000 MWe power plant producing approximately 2000 MWt of thermal energy [6]. Since the cooling system is modular, this unit cost is assumed to apply independently of the overall system rating, and it is escalated at 6.2% per year to obtain a unit cost of $25,800 per MWt of capacity in 1985. Table 6.12 summarizes the resulting cooling tower costs for each of the three system options.

6.4.5 Total Thermal Utility System Costs

Figure 6.8 illustrates the thermal utility system present-worth costs in 1985 as a function of the measured average thermal/electrical load split over the range of system options studied.

TABLE 6.12

AUXILIARY COOLING TOWER COSTS

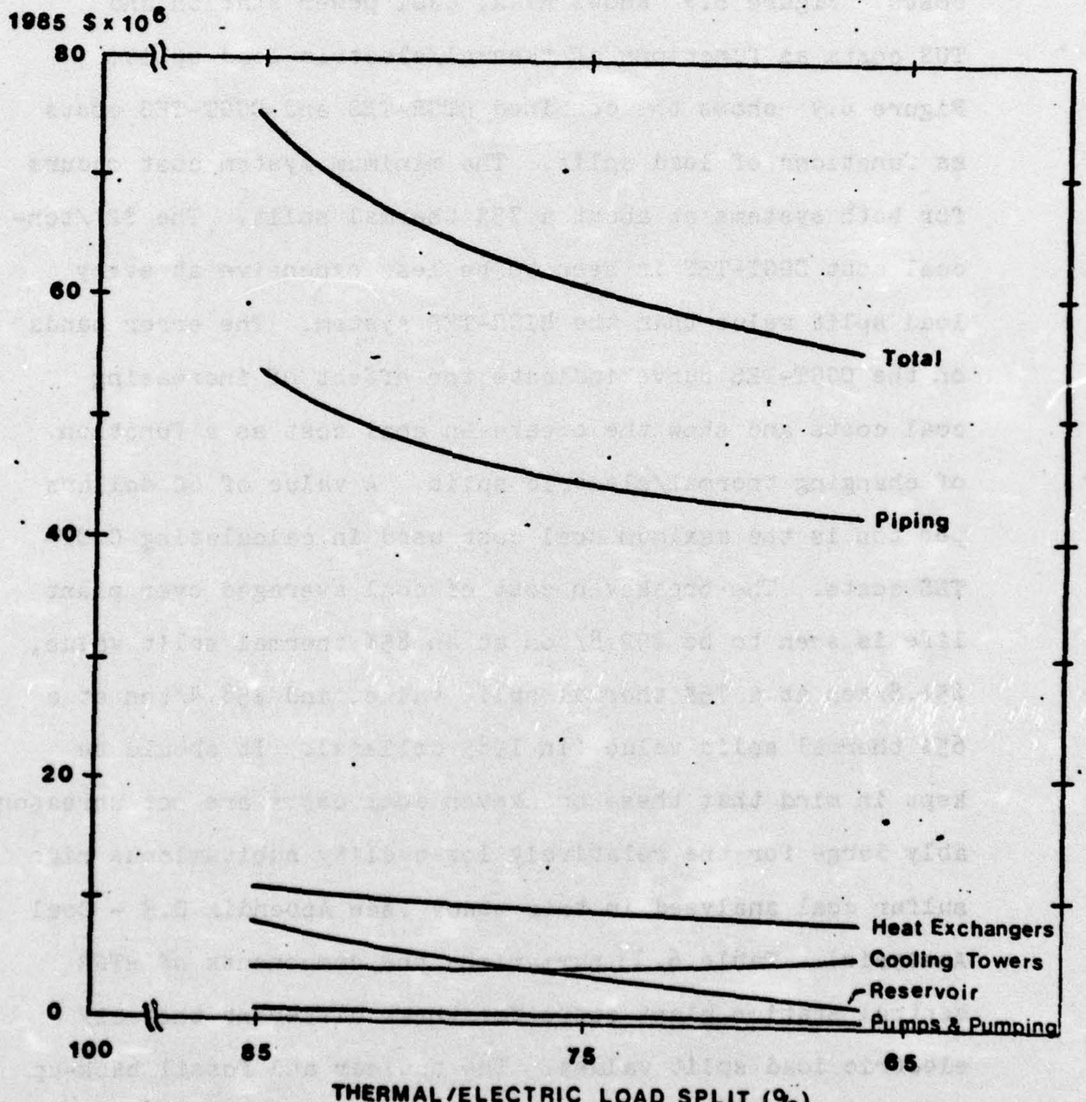
Thermal/Electrical Load Split Option	Cooling Tower Rating[1] (MWt)	1985 Cost[2] ($ x 10^6)
85%	157	4.05
75%	150	3.87
65%	179	4.61

(1) Assumed use of a set of mechanical draft units to meet the peak system thermal power output.

(2) Base cost = $25,800/MWt.

Figure 6.8

Thermal Utility System Cost
as a Function of
Thermal/Electrical Load Split Value



6.5 TES Cost Minimization

The TES cost is found by adding the costs of either the CGGT or HTGR power station to the appropriate TUS costs. Figure 6.9 shows HTGR, CGGT power station and TUS costs as functions of thermal/electric load split; Figure 6.9 shows the combined HTGR-TES and CGGT-TES costs as functions of load split. The minimum system cost occurs for both systems at about a 75% thermal split. The $27/ton-coal cost CGGT-TES is seen to be less expensive at every load split value than the HTGR-TES system. The error bands on the CGGT-TES curve indicate the effect of increasing coal costs and show the breakeven coal cost as a function of changing thermal/electric split. A value of 60 dollars per ton is the maximum coal cost used in calculating CGGT-TES costs. The breakeven cost of coal averaged over plant life is seen to be $52.8/ton at an 85% thermal split value, $51.8/ton at a 75% thermal split value, and $58.4/ton at a 65% thermal split value (in 1985 dollars). It should be kept in mind that these breakeven coal costs are not unreasonably large for the relatively low-quality subituminous high sulfur coal analyzed in this study (see Appendix D.3 - Coal Analysis). Table 6.13 summarizes the components of HTGR central station plant costs for three different thermal/electric load split values. The nuclear and fossil back-up capital costs are shown as well as the fuel, operation and maintenance costs. Table 6.14 shows a similar cost schedule

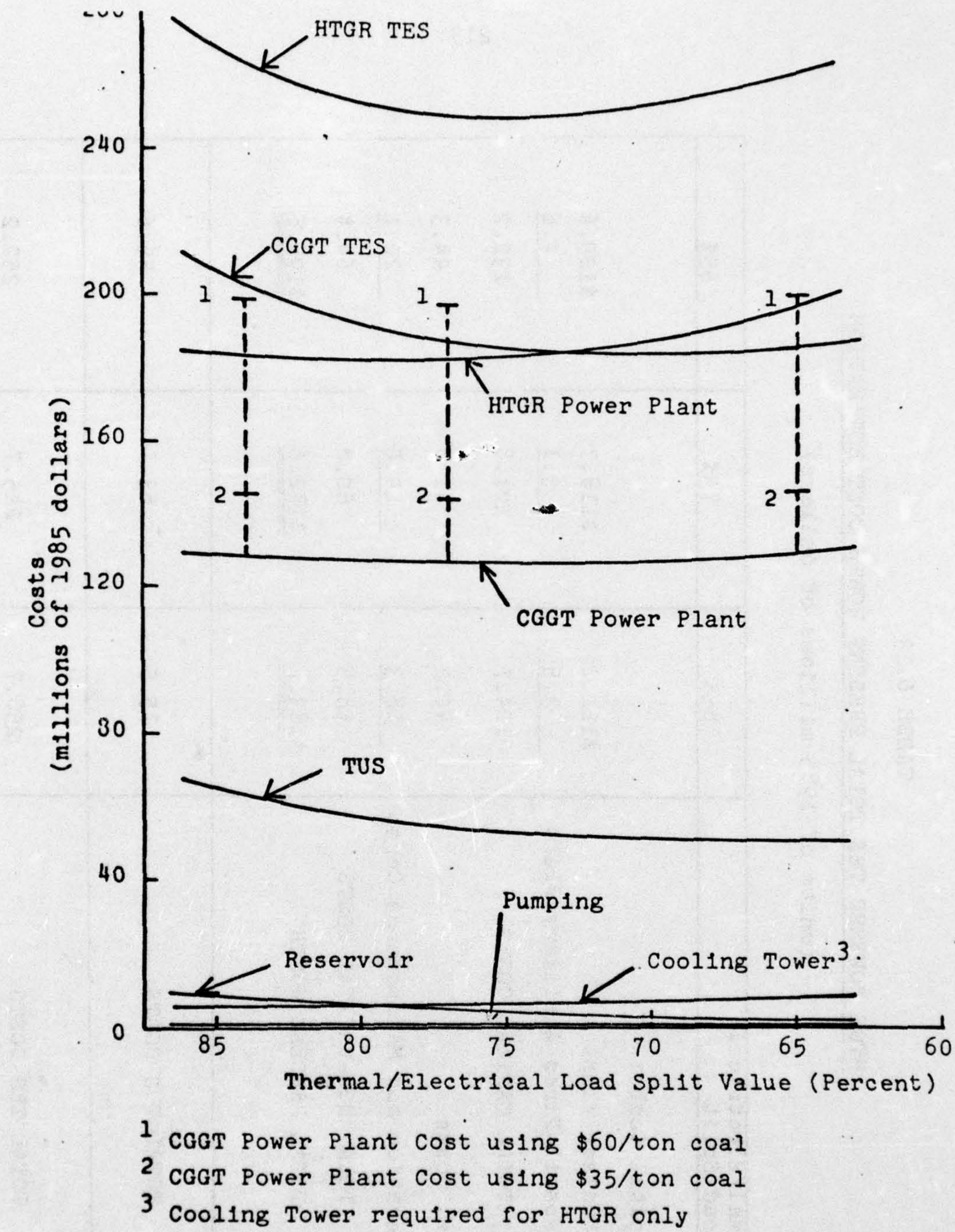


Figure 6.9 Present Worth (1985 Dollars) of Power Plants, Thermal Utility Systems, and combined Total Energy Systems as a function of Thermal/Electrical Load Split Value

TABLE 6.13

HTGR POWERED TES TOTAL PRESENT WORTH COST ESTIMATION

(units of 1985 millions of dollars)

Thermal/Electrical Load Split	85%	75%	65%
Capital Costs:			
Nuclear Plant	$121.2	$118.7	$129.6
Fossil-Fired Auxiliary Plant	3.5	3.1	2.6
TOTAL CAPITAL COSTS	124.7	121.8	132.2
Fuel Costs	40.2	41.8	44.3
Operation and Maintenance Costs	18.3	19.0	20.1
TOTAL NON-CAPITAL COSTS	58.5	60.8	64.4
TOTAL PRESENT WORTH	$183.2	$182.6	$196.6
TOTAL TUS COSTS	75.5	63.1	55.6
TOTAL TES COSTS	258.7	245.7	252.2

TABLE 6.14

CGGT POWERED TES TOTAL PRESENT WORTH COST ESTIMATION

(units of 1985 millions of dollars)

Thermal/Electrical Load Split	85%	75%	65%
Capital Costs	\$ 56.2	\$ 55.3	\$ 58.6
Fuel Cost (at \$27/ton)	57.0	56.8	56.8
Operation and Maintenance Costs	15.4	15.3	15.3
TOTAL PRESENT WORTH COSTS	\$128.6	\$127.4	\$130.7
TOTAL TUS COSTS	71.4	59.2	51.0
TOTAL TES COSTS	200.0	186.6	181.7

for the CGGT plant at various load split values. Perversely, the highest cost item for either system (i.e. HTGR capital cost or CGGT coal cost) is also the item with the greatest cost uncertainty. Experience in building small HTGRs is very limited, and this could lead to large cost overruns should the projected system be constructed. Similarly unpredictable are the costs of coal projected over the 30 year plant lifetime. However, it is felt that the economic analyses presented in this report are based on reasonable assumptions and should provide useful estimates of the anticipated costs.

6.6 Hybrid TES

In the following section, several hybrid TES designs - consisting of combinations of base-loaded nuclear plants with peaking coal gasification - gas turbine plants are analyzed. Additionally the equivalent cost of gas required to fuel the 75% TUS is presented.

Three different conditions of HTGR "base loading" are considered. The first "base loaded" HTGR considered is one whose electrical and thermal output is used completely throughout the year; that is, the plant would operate at 100% power all the time. The yearly minimum in thermal demand for Ft. Bragg is about 15 MW(t). The 100% base loaded HTGR therefore is sized to produce 7.5 MW(e) and 15 MW(t). A complementary

CGGT plant produces the remainder of the electrical and thermal energy demanded. The present worth cost of this hybrid system is $167.8 million in 1985 dollars. The cost breakdown is as shown in Table 6.15 . It is seen that the high capital cost of the HTGR and CGGT do not offset the reduction in CGGT fuel cost which results from combining the HTGR with the CGGT plant [CGGT fuel costs alone are $56.8 million]. Therefore, there exists no incentive to use Hybrid I Option.

The second hybrid system considered is one whose HTGR electrical output is always utilized, with the waste heat going to the 75% thermal split TUS or to a cooling tower as the thermal demand requires. The minimum electric demand occurs on the winter-spring day and is 23.1 MW(e). The HTGR is sized to supply 23.1 MW(e). The CGGT system supplies all peaking thermal and electric loads. Hybrid Option II has a present worth cost of $162.8 million in 1985 dollars as is summarized in Table 6.16. The large reduction in coal costs (from $56.8 to $11.0) is nearly able to offset the increased (from Option I) capital costs of the HTGR. However, it is seen that the costs of a completely fossil-fired CCGT system are lower than those of Hybrid Option II.

The final hybrid option considered is that of an HTGR sized to meet the peak load of the average winter day, with

TABLE 6.15

HYBRID NUCLEAR-FOSSIL POWER PLANT PRESENT WORTH COST ESTIMATES

HYBRID TYPE I

(7.5 MW(e) HTGR and 65% TUS thermal/electrical load split)
(units of 1985 millions of dollars)

HTGR Costs		
7.5 MW(e) Power Plant Capital Costs	$33.5	
Fuel and Operation and Maintenance Costs (20 mills/KW-hr)	0.8	
TOTAL HTGR PRESENT WORTH COSTS	$34.3	million 1985
CGGT Costs		
Power Plant Gasifiers and Turbines Costs	$ 71.7	
Coal Costs ($27/ton)	46.5	
Operation and Maintenance Costs	15.3	
TOTAL CGGT PRESENT WORTH COSTS	$133.5	
TOTAL CENTRAL PLANT PRESENT WORTH COSTS	$167.8	million 1985

TABLE 6.16

HYBRID NUCLEAR-FOSSIL POWER PLANT PRESENT WORTH COST ESTIMATES

HYBRID TYPE II

(23.1 MW(e) HTGR and 75% TUS thermal/electrical load split)
(units of 1985 millions of dollars)

HTGR Costs		
23.1 MW(e) Power Plant Capital Costs	$ 77.8	
Fuel (6.1 mills/KW-hr) and Operation and Maintenance Costs (7.9 mills/KW-hr)	32.1	
TOTAL HTGR PRESENT WORTH COSTS	$109.9	million 198
CGGT Costs		
Power Plant Gasifiers and Turbines Costs	$ 38.1	
Coal Costs	11.0	
Operation and Maintenance Costs	3.8	
TOTAL CGGT PRESENT WORTH COSTS	$ 52.9	
TOTAL CENTRAL PLANT PRESENT WORTH COSTS	$162.8	million 198

design day peaking supplied by the CGGT power plant. Because the CGGT unit supplies only design peaking demands there is very little coal consumption. The associated CGGT fuel costs are assumed to be negligible. The sum of $5 million is allotted for fuel, operation and maintenance of the CGGT plant. The 75% thermal split TUS HTGR is sized to meet the average winter electrical demand peak of 42.1 MW(e). The 75% thermal split TUS winter design day (yearly peak electrical output) is 49.6 MW(e). The CGGT plant must, therefore, supply 7.5 MW(e) on the winter design day. Table 6.17 summarizes the cost components of the Type III Hybrid Option. The complete central plant costs $200.2 million 1985 dollars. The high cost of the Type III hybrid is due chiefly to the low capacity factor (69%) of the HTGR causing high fuel and capital costs. It should be noted that completely eliminating the CGGT peaking system results in an HTGR system, unable to meet the winter peak design day load, and $42.8 million (in 1985 dollars) more expensive than a correctly sized CGGT plant.

The analyses of the stand-alone HTGR power plant and of the hybrid power plants shows that the CGGT central station is always the least expensive option. The equivalent cost of the gas produced by the gasification plant (in 1985 dollars) is 253 ¢/MBTU using $27/ton coal or 398 ¢/MBTU using $50/ton coal. The recommended TES consists of a 75% thermal/electric

TABLE 6.17

HYBRID NUCLEAR-FOSSIL POWER PLANT PRESENT WORTH COST ESTIMATES

HYBRID TYPE III

(42.1 MW(e) HTGR and 75% TUS thermal/electrical load split)
(units of 1985 millions of dollars)

HTGR Costs		
42.1 MW(e) Power Plant Capital Costs	$109.3	
Fuel Costs (17.4 mills/KW-hr)	41.7	
Operation and Maintenance Costs (8.0 mills/KW-hr)	19.2	
TOTAL HTGR PRESENT WORTH COSTS	$170.2	million
CGGT Costs		
Power Plant Gasifiers and Turbines Costs	$ 25.0	
Fuel and Operation and Maintenance Costs	5.0	
TOTAL CGGT PRESENT WORTH COSTS	$ 30.0	
TOTAL CENTRAL PLANT PRESENT WORTH COSTS	$200.0	million

load split TES powered by a CGGT central power plant. Using a 10% cost of money, 30 year plant lifetime, straight line depreciation and all capital and fuel charges being assessed against the electrical energy product, it is seen that the electric energy produced by this central plant costs 58.2 mills/KW-hr.

REFERENCES

1. Metcalfe, L.J., Driscoll, M.J., "Economic Assessment of Nuclear and Fossil-Fired Energy Systems for DOD Installation," Project Report, Contract No, DAAK02-74-C-0308, Department of Nuclear Engineering, M.I.T., February 1975.

2. Kelly, J., Golay, M.W., "Economic and Technical Aspects of Gas Turbine Power Stations in Total Energy Applications," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, M.I.T., 1976.

3. Boyd, W.C., Golay, M.W., "Economic and Technical Aspects of Coal Gasification for use in Gas Turbine Operation," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, M.I.T., 1976.

4. "Preliminary Study - A Total Energy System for Boston," MIT Department of Nuclear Engineering Subject 22.33 Final Report, May 1976. Original source of data: "The U.S. Energy Problem," Vol. II, Appendices - Part A - Inter Technology, Nov. 1972.

5. Nida, A.V., "Nuclear Total Utility System for Military Installations," MIT Department of Nuclear Engineering, S.M. Thesis, Jan. 1974.

6. Personal communication, Prof. M.W. Golay, MIT Department of Nuclear Engineering, July 1976.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

7.1 Conclusions

From the economic analysis of Chapter 6 it is seen that the minimum cost TES for thermal/electric load split values between 65% and 100% occurs at a split value of 75%. The minimum cost TES for both the HTGR and CGGT plants occurs at approximately the same split value. The break-even cost of coal, compared to the HTGR option, varies as a function of the thermal/electric split value. The minimum cost HTGR-TES results in a breakeven coal cost of $51.8/ton in 1985 dollars. Using a coal cost of $27/ton in 1985 dollars, the CGGT power plant electrical generating cost is $58.2 mills/KW-hr if all costs are assigned to the electrical power product. Note that this generating cost includes the capital cost of waste heat exchangers and gasifier capacity sized to meet the thermal rather than the electrical demand.

Figure 7.1 shows the variation of the required HTGR power plant capacity factor for the TES as a function of the thermal/electrical load split value. It is seen that the maximum annually-averaged capacity factor is 59%, and that this occurs at a thermal/electric load split value of approximately 75%. The HTGR sized to supply the 75%

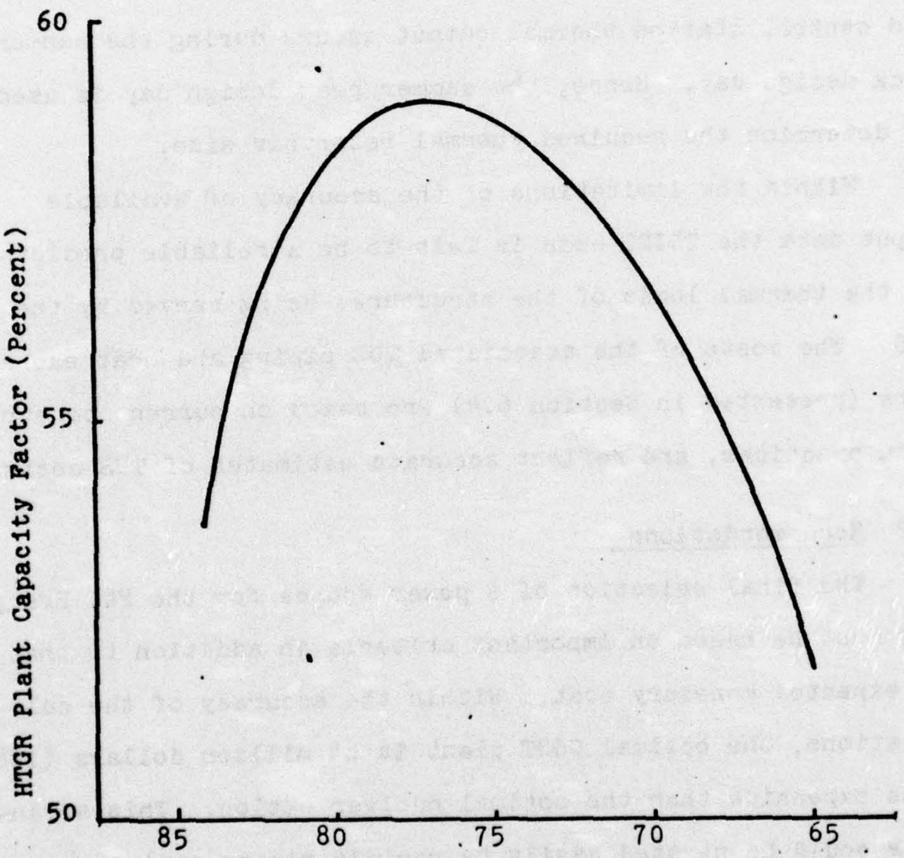


Figure 7.1. HTGR Power Plant Capacity Factor vs. Thermal/Electric Load Split Value

thermal/electric split value TUS, therefore, has the lowest fuel-cost charges as well as the lowest capital costs.

The required power plant capacity for Ft. Bragg is determined by the winter peak design day. However, the largest instantaneous mismatch between TUS thermal demand and central station thermal output occurs during the summer peak design day. Hence, the summer peak design day is used to determine the required thermal reservoir size.

Within the limitations of the accuracy of available input data the TDIST code is felt to be a reliable predictor of the thermal loads of the structures being served by the TES. The costs of the associated TUS piping ahd heat exchangers (presented in Section 6.4) are based on current construction practices, and reflect accurate estimates of TUS costs.

7.2 Recommendations

The final selection of a power source for the Ft. Bragg TES must be based on important criteria in addition to that of expected monetary cost. Within the accuracy of the calculations, the optimal CGGT plant is 64 million dollars (1985) less expensive than the optimal nuclear option. This advantage could be negated easily by rapidly rising coal costs. Alternatively, cost overruns on the HTGR power plant could increase significantly the cost of the nuclear option. In addition, each system has important secondary characteristics which argue in favor of its selection for TES use.

Some of the more important power plant selection trade-offs are summarized in Table 1.1. For example, the CGGT system has the advantage of modular add-on potential growth. It is relatively easy to install another gasifier or gas turbine unit to the power plant as is required by the expansion of the base TES. The HTGR power plant is limited severely in its add-on growth ability.

The HTGR does have a significant advantage in the dependability of its fuel supply. A freshly fueled HTGR would be expected to supply from 3 to 6 years of service before refueling would be required. Conversely, storage of more than a few months' supply of coal on base would be impractical because of the large bulk of such a coal pile.

The ultimate selection of the power plant type must be based on the users present and projected needs not only for electrical and thermal power, but also for personnel with experience in new technologies.

There are several additional recommendations which, while beyond the scope of this report, should be kept in mind for future work. The high cost of the TUS piping, pumps and heat exchangers suggests that alternate technologies such as all electric heat pump heating may be attractive. In supplying such an all-electric utility system it should be noted that gas turbine power plants would be very easy to employ. In the work for this report a steam Rankine bottoming cycle powered by the gas turbine exhaust - cooled previously in the TUS heat exchanger - has been studied. However, because turbine exhaust energy exceeds the TUS demand by a small amount for only a few months of the year the cost of coal saved by steam bottoming is less than the cost of the steam bottoming cycle equipment. Steam bottoming is therefore not discussed in this report. An alternate all-electric system could use the gas turbine exhaust to generate steam, the steam used to drive a high back-pressure turbine, with the steam turbine exhaust (at about 230 °F) supplying power for the TUS. In any case, a more extensive analysis of TUS economics should be performed to establish definitive conditions for the economic use of hot water piping systems.

APPENDIX A.1

FUEL SAVINGS ACHIEVED BY A CENTRAL STATION

TUS COMPARED TO CONVENTIONAL HEATING

Central Station Gas-Fired Heater Efficiency

$$= \frac{\text{BTU absorbed by water}}{\text{BTU of fuel consumed}} = 70\% \quad [6]$$

Individual "Home" Gas-Fired Heater Efficiency

$$= \frac{\text{BTU absorbed by water}}{\text{BTU of fuel consumed}} = 40\% \quad [6]$$

Average fraction of thermal load recovered by Turbine
Exhaust Waste heat exchangers = 85.4% for 75% thermal split

Assume heat load of 100 units

1. "Home" Heaters would require

$$\frac{100}{40\%} = 250 \text{ units}$$

2. Central Station TUS would require

$$\frac{100(1 - .854)}{70\%} = 20.9 \text{ units}$$

Thus the Central Station TUS reduces fuel consumption by

$$\frac{229.1 \text{ units}}{250 \text{ units}} = 91.7\%$$

APPENDIX A.2

COST OF ENERGY STORAGE AS HOT WATER

COMPARED TO GAS STORAGE

I. Gas Storage at 300 psi costs $\$12/ft^3$

For our system gas H.V. = 125 BTU/SCF

Cost of Energy Storage as Gas at 300 psi =

$$\frac{\$12}{ft^3} \; \frac{SCF}{125\ BTU} \; \frac{ft^3}{0.49\ SCF} = \frac{\$1.96}{BTU}$$

II. Hot Water Storage $\$7.5/ft^3$

Energy stored in each ft^3 =

$$\frac{1\ BTU}{lbm\,°F}\,(380°F-150°F)\,\frac{lbm}{.017ft^3} = 13529\ BTU/ft^3$$

Cost of energy storage as hot water =

$$\frac{\$7.5}{ft^3}\left[\frac{1}{13529\ \frac{BTU}{ft^3}}\right] = \frac{\$.00055}{BTU}$$

Clearly it is cheaper to store energy as hot water than gas.

APPENDIX B

FORT BRAGG CONSUMER SPECIFICATIONS

In order to be able to compute the conduction, solar incidence, ventilation and internal heat generation components of the space conditioning demands and the domestic hot water usage for the specified energy consumer catogories, TDIST requires the following data for each building type to be analyzed: the exposed areas and thermal resistances of walls, windows, the roof and the basement; the building height; its orientation; the outer wall and roof surface materials; the wall and roof solar absorptivities; a composite internal room and glass material window shading coefficient; the shading of each wall and the roof; the nominal maximum desired ventilation air flow rate; the total connected electrical load in the building, exclusive of any electrical space conditioning equipment; the maximum rate of domestic hot water usage; crack lengths and flow coefficients for openings around doors and windows and cracks in the structural walls; a desired internal room temperature profile to be maintained by the space conditioning equipment throughout the analysis period; a schedule of building use factors relating appliance, lighting and ventilation requirements to the building occupancy; and a schedule of domestic hot water usage.

Because much of these data depend strongly upon the precise nature of the building types chosen to be analyzed, Army personnel were requested to supply as much of the information as possible for each of the representative Fort Bragg building categories. A continuing effort at data acquisition and analysis by the Army and several preliminary calculations which demonstrated the sensitivity of the building loads to variations in each of these parameters were coordinated to yield revisions to some of the initial building specifications to reflect more accurately the combined effects of building structural characteristics, Army personnel lifestyles and military energy usage and conservation regulations expected to be realized in 1985. In particular, special care was taken in specifying the building shading coefficients, infiltration air flow coefficients, ventilation requirements and building use profiles, since it has been found that solar radiational heating and the combined effects of infiltration and forced air ventilation air flows contribute significantly to the total building space conditioning loads. Large variations in these coefficients in the available literature, a general lack of detailed information from direct field measurements, and the individual building-specific nature of such factors as tree shading and window weatherstripping made the task of formulating these specifications for a "typical" building unit especially difficult. Where possible, results from the TDIST load calculations were compared with documented simulations or measured energy usage data, but the specific nature of the results, the large variability among seemingly similar building types and a lack of detailed parameter identification led to marginal success in verifying the TDIST analysis for only a residential and a commercial building unit. Therefore, because of this general lack of documented information for verification of either the input data or the results of the Fort Bragg simulations, special care was taken to insure that all building specifications were consistent with available measurements, and, where directly measured data was unavailable or where the aggregate nature of the representative units made specific measurements inapplicable, a cross-section of suggested design values, data used for published building simulations and information based upon Army personnel observations and experience were used to generate "reasonable" input parameters believed to accurately reflect the average conditions found at Fort Bragg. Tables B.1-B.11 present the final data specifications and assumptions made for each of the eleven consumer categories.

References

1. "Service Water Heating," Chapter 37, ASHRAE Handbook and Product Directory, 1973 Systems, American Society of Heating, Refrigerating and Air Conditioning Engineers, New York, 1973, pp. 37.10-37.19.

2. Kusuda, T., "INFIL, An Algorithm for Calculating Air Infiltration," NBSLD, Computer Program for Heating and Cooling Loads in Buildings, NBSIR-74-574, National Bureau of Standards, Washington, D.C., Nov. 1974, pp. 129a-143a.

TABLE B.1

BUILDING INPUT DATA

Type 1: Troop Housing: Modern

Unit: Single Residence Module

Wall Area: 10,368 ft^2

Wall Composite Thermal Resistance: 3.70 (hr)(ft^2)(°F)/BTU

Window Area: 4110 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
- Assumed single pane, no storm windows

Roof Area: 2060 ft^2

Roof Composite Thermal Resistance: 14,29(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 2060 ft^2

Basement Wall Thermal Resistance: 10.00 (hr)(ft^2)(°F)/BTU

Building Height: 27.5 ft.

Building Orientation: 45° from North

Wall Surface Material: Stucco

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.50

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.50 (50% of incident radiation transmitted)
- Assumed use of blinds, shades or drapes as in typical residences

Wall Fraction Lit: 0.80 (20% of each wall shaded)
- Assumed, based on photographs of typical residences

Roof Fraction Lit: 1.00 (no shading)

TABLE B.1 (continued)

Door Crack Length: 78 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
- See Table B.12

Window Crack Length: 800 ft.

Window Air Flow Coefficients: C: 1.7 N: 0.66
- See Table B.12

Wall Air Flow Coefficients: C: 0.004 N: 0.70
- See Table B.12

Peak Ventilation: 944 CFM
- Assumed, one air change per hour as per Army measurements and typical residentail data

Connected Electrical Load: 5.37 KW
- Primarily lighting at 0.87 watts/ ft^2 total floor area

Peak Domestic Hot Water Demand: 129,856 BTU/hr
- Assumed peak of 3.8 gph per person, 63 residents, 239.4 gal/hr total maximum as per ASHRAE Systems, 1973.[1]

Winter Room Temperatures: 68 °F (minimum)

Summer Room Temperatures: 78 °F (maximum)

TABLE B.1 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from Ref.1 for Dormitories)
12		.81	.33
1		.67	.26
2		.61	.13
3		.58	.11
4	AM	.52	.03
5		.49	.04
6		.52	.01
7		.59	.11
8		.66	.18
9		.69	.21
10		.79	.22
11		.90	.18
12		.93	.24
1		.96	.16
2		.96	.13
3		.93	.16
4	PM	.95	.24
5		.93	.20
6		.98	.26
7		1.00	.30
8		.99	.29
9		.96	.16
10		.93	.26
11		.87	.37
12		.81	.33

TABLE B.2

BUILDING HEAT DATA

Type 2: Family Housing: 3648

Unit: Two Family Duplex

Wall Area: 4784 ft^2

Wall Composite Thermal Resistance: 10.00 (hr)(ft^2)(°F)/BTU

Window Area: 330 ft^2

Window Thermal Resistance: 0.89 (hr)(ft^2)(°F)/BTU
- Assumed single pane, no storm windows

Roof Area: 1519 ft^2

Roof Composite Thermal Resistance: 16.67 (hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 1824 ft^2

Basement Wall Thermal Resistance: 4.91(hr)(ft^2)(°F)/BTU

Building Height: 23 ft.

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.45 (45% of incident radiation transmitted)
- Assumed blinds or drapes as in typical residences

Wall Fraction Lit: 0.70 (30% of each wall shaded)
- Assumed, based on photographs of Fort Bragg housing

TABLE B.2 (continued)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 31 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
·See Table B.12

Window Crack Length: 195 ft.

Window Air Flow Coefficients: C: 1.7 N: 0.66
·See Table B.12

Wall Air Flow Coefficients: C: 0.01 N: 0.80
·See Table B.12

Peak Ventilation: 702 CFM
·Assumed one air change per hour as per Army measurements and typical residential data

Connected Electrical Load: 3.17 KW
·Primarily lighting at 0.87 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: 13,018 BTU/hr
·Assumed peak of 24 gal/hr to serve two families as per ASHRAE Systems, 1973[1]

Winter Room Temperatures: 72 °F (minimum)

Summer Room Temperatures: 75 °F (maximum)

TABLE B.2 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from Ref.1 for Apartments)
12		.81	.17
1		.67	.14
2		.61	.13
3		.58	.10
4	AM	.52	.11
5		.49	.10
6		.52	.10
7		.59	.13
8		.66	.15
9		.69	.25
10		.79	.21
11		.90	.19
12		.93	.17
1		.96	.18
2		.96	.15
3		.93	.13
4	PM	.95	.12
5		.93	.12
6		.98	.15
7		1.00	.19
8		.99	.21
9		.96	.18
10		.93	.15
11		.87	.13
12		.81	.17

TABLE B.3

BUILDING INPUT DATA

Type 3: Family Housing: Row

Unit: Four Family Dwelling

Wall Area: 5250 ft^2

Wall Composite Thermal Resistance: 4.00(hr)(ft^2)(°F)/BTU

Window Area: 532 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
•Assumed single pane, no storm windows

Roof Area: 3664 ft^2

Roof Composite Thermal Resistence: 4.55(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 3750 ft^2

Basement Wall Thermal Resistence: 4.91(hr)(ft^2)(°F)/BTU

Building Height: 17.5 ft.

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.50(50% of incident radiation transmitted)
•Assumed use of blinds or drapes as in typical residences

Wall Fraction Lit: 0.90 (10% of each wall shaded)
•Assumed, based on photographs of Fort Bragg Housing

Roof Fraction Lit: 1.00 (no shading)

TABLE B.3 (continued)

Door Crack Length: 132 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
•See Table B.12

Window Crack Length: 306 ft.

Window Air Flow Coefficients: C: 1.7 N: 0.66
•See Table B.12

Wall Air Flow Coefficients: C: 0.004 N: 0.70
•See Table B.12

Peak Ventilation: 1103 CFM
•Assumed one air change per hour as per Army measurements and typical residential data

Connected Electrical Load: 6.53 KW
•Primarily lighting at 0.87 watts/ ft^2 total floor area

Peak Domestic Hot Water Demand: 39,055 BTU/hr
•Assumed peak of 72 gal/hr to serve up to six families as per ASHRAE Systems, 1973[1]

Winter Room Temperatures: 72 °F (minimum)

Summer Room Temperatures: 75 °F (maximum)

TABLE B.3 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from Ref.1 for Apartments)
12		.81	.17
1		.67	.14
2		.61	.13
3		.58	.10
4	AM	.52	.11
5		.49	.10
6		.52	.10
7		.59	.13
8		.66	.15
9		.69	.25
10		.79	.21
11		.90	.19
12		.93	.17
1		.96	.18
2		.96	.15
3		.93	.13
4	PM	.95	.12
5		.93	.12
6		.98	.15
7		1.00	.19
8		.99	.21
9		.96	.18
10		.93	.15
11		.87	.13
12		.81	.17

TABLE B.4

BUILDING INPUT DATA

Type 4: Family Housing: 34

Unit: Single Family Detached Dwelling

Wall Area: 5244 ft^2

Wall Compsite Thermal Resistance: 3.45(hr)(ft^2)(°F)/BTU

Window Area: 904 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
•Assumed Single pane, no storm windows

Roof Area: 2325 ft^2

Roof Composite Thermal Resistance: 16.67(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 2074 ft^2

Basement Wall Thermal Resistance: 4.91(hr)(ft^2)(°F)/BTU

Building Height: 28.5 ft

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.50(50% of incident radiation transmitted)
•Assumed use of blinds or drapes as in typical residences

Wall Fraction Lit: 0.90(10% of each wall shaded)
•Assumed, based on photographs of Fort Bragg housing

TABLE B.4 (continued)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 30 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
- See Table B.12

Window Crack Length: 296 ft.

Window Air Flow Coefficients: C: 1.7 N: 0.66
- See Table B.12

Wall Air Flow Coefficients: C: 0.01 N: 0.80
- See Table B.12

Peak Ventilation: 987 CFM
- Assumed one air change per hour as per Army measurements and typical residential data

Connected Electrical Load: 2.08 KW
- Primarily lighting at 0.50 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: 6509 BTU/hr
- Assumed peak of 12 gal/hr for a family of four as per ASHRAE Systems, 1973 [1]

Winter Room Temperature: 72 °F (minimum)

Summer Room Temperature: 75 °F (maximum)

TABLE B.4 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from Ref.1 for Apartments)
12		.81	.17
1		.67	.14
2		.61	.13
3		.58	.10
4	AM	.52	.11
5		.49	.10
6		.52	.10
7		.59	.13
8		.66	.15
9		.69	.25
10		.79	.12
11		.90	.19
12		.93	.17
1		.96	.18
2		.96	.15
3		.93	.13
4	PM	.95	.12
5		.93	.12
6		.98	.15
7		1.00	.19
8		.99	.21
9		.96	.18
10		.93	.15
11		.87	.13
12		.81	.17

TABLE B.5

BUILDING INPUT DATA

Type 5: Fort Bragg Hospital

Unit: Hospital

Wall Area: 73,000 ft^2

Wall Composite Thermal Resistance: 3.70(hr)(ft^2)(°F)/BTU

Window Area: 24,333 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
- Assumed single pane, no storm windows

Roof Area: 60,000 ft^2

Roof Composite Thermal Resistance: 12.50(hr)(ft^2(°F)/BTU

Basement Ground-Contact Area: 60,000 ft^2

Basement Wall Thermal Resistance: 6.50(hr)(ft^2)(°F)/BTU

Building Height: 90 ft

Building Orientation: North

Wall Surface Material: Concrete Block

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.91

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.40(40% of incident radiation transmitted)
- Assumed use of shades as shown in hospital photographs

Wall Fraction Lit: 0.95 (5% of each wall shaded)
- Assumed, based on hospital photographs

TABLE B.5 (continued)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 230 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
•See Table B.12

Window Crack Length: 12,167 ft.

Window Air Flow Coefficients: C: 3.2 N: 0.66
•See Table B.12

Wall Air Flow Coefficients: C: 0.004 N: 0.70
•See Table B.12

Peak Ventilation: 270,033 CFM
•Assumed three air changes per hour as average of Army recommendations for various areas in hospital ranging from one to twelve changes per hour

Connected Electrical Load: 337.06 KW
•Average of total demand given as 0.82 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: 1,152,653 BTU/hr
•Assumed peak of 4.25 gal/hr per bed, 500 beds, as per ASHRAE Systems, 1973[1]

Winter Room Temperatures: 74 °F (constant)

Summer Room Temperatures: 74 °F (constant)

TABLE B.5 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (assumed to vary with building use)
12		.53	.53
1		.41	.14
2		.41	.41
3		.41	.41
4	AM	.41	.41
5		.38	.38
6		.53	.53
7		.60	.60
8		.71	.71
9		.88	.88
10		.94	.94
11		.98	.98
12		.99	.99
1		1.00	1.00
2		1.00	1.00
3		1.00	1.00
4	PM	.99	.99
5		.93	.93
6		.79	.79
7		.70	.70
8		.70	.70
9		.68	.68
10		.59	.59
11		.56	.56
12		.53	.53

TABLE B.6

BUILDING INPUT DATA

Type 6: Storage

Unit: Warehouse

Wall Area: 7104 ft^2

Wall Composite Thermal Resistance: 3.70(hr)(ft^2)(°F)/BTU

Window Area: 1420 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
·Assumed single pane, no storm windows

Roof Area: 1,421 ft^2

Roof Composite Thermal Resistance: 3.33(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 11,421 ft^2

Basement Wall Thermal Resistance: 6.00(hr)(ft^2)(°F)/BTU

Building Height: 15.75 ft.

Building Orientation: 45° from North

Wall Surface Material: Concrete Block

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.68

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.80(80% of incident radiation transmitted)
·Assumed some windows dirty or blocked by stored goods

Wall Fraction Lit: 1.00 (no shading)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 176 ft.

TABLE B.6 (continued)

Door Air Flow Coefficients: C: 40 N: 0.50
·See Table B.12

Window Crack Length: 222 ft.

Window Air Flow Coefficients: C: 2.2 N: 0.66
·See Table B.12

Wall Air Flow Coefficients: C: 0.01 N: 0.80
·See Table B.12

Peak Ventilation: 3001 CFM
·Assumed one air change per hour as conservative requirement

Connected Electrical Load: 22.84 KW
·Primarily lighting at 2.0 watts/ ft^2 total floor area as per Army specifications

Peak Domestic Hot Water Demand: Negligible

Winter Room Temperatures: 65 °F (minimum)

Summer Room Temperatures: Not air conditioned

TABLE B.6 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)
12		.10
1		.10
2		.10
3		.10
4	AM	.10
5		.10
6		.10
7		.30
8		.30
9		.95
10		.95
11		.95
12		.95
1		.95
2		.95
3		.95
4	PM	.95
5		.50
6		.10
7		.10
8		.10
9		.10
10		.10
11		.10
12		.10

TABLE B.7

BUILDING INPUT DATA

Type 7: Community

Unit: Recreation/Community Center

Wall Area: 10,556 ft^2

Wall Composite Thermal Resistance: 3.13(hr)(ft^2)(°F)/BTU

Window Area: 300 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
·Assumed single pane, no storm windows

Roof Area: 20,369 ft^2

Roof Composite Thermal Resistance: 5.89(hr)(ft^2(°F)/BTU

Basement Ground-Contact Area: 20,486 ft^2

Basement Wall Thermal Resistance: 5.27(hr)(ft^2)(°F)/BTU

Building Height: 15 ft.

Building Orientation: 45° from North

Wall Surface Material: Concrete Block

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.68

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.70(70% of incident radiation transmitted)
·Assumed, based upon photograph of community center

Wall Fraction Lit: 1.00 (no shading)

Roof Fraction Lit: 1.00 (no shading)

TABLE B.7 (continued)

Door Crack Length: 66 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
·See Table B.12

Window Crack Length: 150 ft.

Window Air Flow Coefficients: C: 3.0 N: 0.66
·See Table B.12

Wall Air Flow Coefficients: C: 0.004 N: 0.80
·See Table B.12

Peak Ventilation: 25,624 CFM
·Assumed five air changer per hour as per Army measurements

Connected Electrical Load: 30.73 KW
·Primarily lighting at 1.5 watts/ ft^2 total floor area

Peak Domestic Hot Water Demand: Unavailable

Winter Room Temperatures: 70 °F (minimum)

Summer Room Temperatures: 78 °F (maximum)

TABLE B.7 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)
12		.15
1		.15
2		.15
3		.15
4	AM	.15
5		.15
6		.15
7		.15
8		.15
9		.25
10		.50
11		1.00
12		1.00
1		1.00
2		1.00
3		1.00
4	PM	1.00
5		1.00
6		1.00
7		1.00
8		1.00
9		.90
10		.90
11		.80
12		.15

TABLE B 8

BUILDING INPUT DATA

Type 8: Administration and Training

Unit: Training Building

Wall Area: 14,782 ft^2

Wall Composite Thermal Resistance: 4.00(hr)(ft^2)(°F)/BTU

Window Area: 5666 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)°F)/BTU
•Assumed single pane, no storm windows

Roof Area: 8135 ft^2

Roof Composite Thermal Resistance: 20.00(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 8.35 ft^2

Basement Wall Thermal Resistance: 4.04(hr)(ft^2)(°F)/BTU

Building Height: 23 ft.

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Slate

Wall Solar Absorptivity: 0.91

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.60(60% of incident radiation transmitted)
•Assumed use of shades as shown in training building photograph

Wall Fraction Lit: 0.90(10% of each wall shaded)
•Assumed, based on training building photograph

TABLE B.8 (continued)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 90 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
·See Table B.12

Window Crack Length: 1304 ft.

Window Air Flow Coefficients: C: 3.2 N: 0.66
·See Table B.12

Wall Air Flow Coefficients: C: 0.004 N: 0.80
·See Table B.12

Peak Ventilation: 7813 CFM
·Assumed 2.5 air changes per hour as per Army measurements and typical office building data

Connected Electrical Load: 72.34 KW
·Primarily lighting at 3 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: Negligible

Winter Room Temperatures: 70 °F (minimum)

Summer Room Temperatures: 75 °F (maximum)

TABLE B.8 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)
12		.05
1		.05
2		.05
3		.05
4	AM	.05
5		.05
6		.10
7		.30
8		.80
9		.95
10		.95
11		.95
12		.75
1		.75
2		.95
3		.95
4	PM	.50
5		.30
6		.10
7		.10
8		.10
9		.10
10		.05
11		.05
12		.05

TABLE B.9

BUILDING INPUT DATA

Type 9: Operations and Maintenance

Unit: Machine Shop

Wall Area: 17,800 ft^2

Wall Composite Thermal Resistance: 2.63(hr)(ft^2)(°F)/BTU

Window Area: 3560 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
·Assumed single pane, no storm windows

Roof Area: 41,850 ft^2

Roof Composite Thermal Resistance: 5.26(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 41,850 ft^2

Basement Wall Thermal Resistance: 4.00(hr)(ft^2)(°F)/BTU

Building Height: 20 ft.

Building Orientation: 45° from North

Wall Surface Material: Concrete Block

Roof Surface Material: Asphalt shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.80(80% of incident radiation transmitted
·Assumed windows generally dirty and partly obstructed

Wall Fraction Lit: 1.00 (no shading)

Roof Fraction Lit: 1.00 (no shading)

TABLE B.9 (continued)

Door Crack Length: 158 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
- See Table B.12

Window Crack Length: 3816 ft.

Window Air Flow Coefficients: C: 3.2 N: 0.66
- See Table B.12

Wall Air Flow Coefficients: C: 0.004 N: 0.80
- See Table B.12

Peak Ventilation: 13,958 CFM
- Assumed one air change per hour as recommended for light manufacturing facilities

Connected Electrical Load: 41.85 KW
- Primarily lighting at 1.0 watt/ ft^2 total floor area

Peak Domestic Hot Water Demand: Negligible

Winter Room Temperature: 65 °F (minimum)

Summer Room Temperatures: Not air conditioned

TABLE B.9 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)
12		.15
1		.10
2		.10
3		.10
4	AM	.10
5		.10
6		.10
7		.10
8		.50
9		.75
10		.80
11		1.00
12		.95
1		.95
2		.90
3		.90
4	PM	.75
5		.75
6		.35
7		.15
8		.15
9		.15
10		.15
11		.15
12		.15

TABLE B.10

BUILDING INPUT DATA

Type 10: Troop Housing: Brick

Unit: Barracks Unit

Wall Area: 25,598 ft^2

Wall Composite Thermal Resistance: 2.63(hr)(ft^2)(°F)/BTU

Window Area: 5261 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
- Assumed single pane, no storm windows

Roof Area: 18,685 ft^2

Roof Composite Thermal Resistance: 5.26(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 18,585 ft^2

Basement Wall Thermal Resistance: 4.00(hr)(ft^2)(°F)/BTU

Building Height: 28.5 ft

Building Orientation: 45° from North

Wall Surface Material: Brick

Roof Surface Material: Asphalt shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.50(50% of incident radiation transmitted)
- Assumed use of shades as in typical residences

Wall Fraction Lit: 0.80 (20% of each wall shaded)
- Assumed, based on photographs of typical residences

TABLE B.10 (continued)

Roof Fraction Lit: 1.00 (no shading)

Door Crack Length: 84 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
•See Table B.12

Window Crack Length: 2368 ft.

Window Air Flow Coefficients: C: 3.2 N: 0.66
•See Table B.12

Wall Air Flow Coefficients: C: 0.004 N: 0.80
•See Table B.12

Peak Ventilation: 13,397 CFM
•Assumed 1.5 air changes per hour as per Army measurements and typical residential data

Connected Electrical Load: 127.40 KW
•Primarily lighting at 2.5 watts/ ft^2 total floor area

Peak Domestic Hot Water Demand: 430,100 BTU/hr
•Assumed peak of 3.8 gph per person; 200 residents, as per ASHRAE Systems, 1973[1]

Winter Room Temperatures: 68 °F (minimum)

Summer Room Temperatures: 78 °F (maximum)

TABLE B.10 (continued)

Time		Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from Ref.1 for Dormitories)
12		.20	.33
1		.20	.26
2		.20	.18
3		.20	.11
4	AM	.20	.03
5		.20	.04
6		.50	.01
7		.90	.11
8		.30	.18
9		.30	.21
10		.30	.22
11		.50	.18
12		.80	.24
1		.50	.16
2		.30	.13
3		.30	.16
4	PM	.50	.24
5		.70	.20
6		.80	.26
7		.80	.30
8		.80	.29
9		.80	.16
10		.50	.26
11		.20	.37
12		.20	.33

TABLE B.11

BUILDING INPUT DATA

Type 11: Troop Housing: Block

Unit: Barracks Unit

Wall Area: 20,590 ft^2

Wall Composite Thermal Resistance: 2.44(hr)(ft^2)(°F)/BTU

Window Area: 5433 ft^2

Window Thermal Resistance: 0.89(hr)(ft^2)(°F)/BTU
•Assumed single pane, no storm windows

Roof Area: 17,000 ft^2

Roof Composite Thermal Resistance: 5.26(hr)(ft^2)(°F)/BTU

Basement Ground-Contact Area: 17,000 ft^2

Basement Wall Theraml Resistance: 4.50(hr)(ft^2)(°F)/BTU

Building Height: 30 ft.

Building Orientation: 45° from North

Wall Surface Materal: Concrete Block

Roof Surface Material: Asphalt Shingle

Wall Solar Absorptivity: 0.70

Roof Solar Absorptivity: 0.80

Window Shading Coefficient: 0.60 (60% of incident radiation transmitted)
•Assumed use of shades as in typical residences

Wall Fraction Lit: 1.00 (no shading)

Roof Fraction Lit: 1.00 (no shading)

TABLE B.11 (continued)

Door Crack Length: 88 ft.

Door Air Flow Coefficients: C: 40 N: 0.50
•See Table B.12

Window Crack Length: 2264 ft.

Window Air Flow Coefficients: C: 3.2 N: 0.66
•See Table B.12)

Wall Air Flow Coefficients: C: 0.004 N: 0.80
•See Table B.12

Peak Ventilation: 12,818 CFM
•Assumed 1.5 air changes per hour as per Army measurements and typical residential data

Connected Electrical Load: 127.40 KW
•Primarily lighting at 2.5 watts/ft^2 total floor area

Peak Domestic Hot Water Demand: 344,932 BTU/hr
•Assumed peak of 3.8 gph per person, 160 residents, as per ASHRAE Systems, 1973 [1]

Winter Room Temperatures: 70 °F (minimum)

Summer Room Temperatures: 78 °F (maximum)

TABLE B.11 (continued)

Time	Building Use Factor (for electrical equipment and ventilation)	Domestic Hot Water Use Factor (from Ref.1 for Dormitories)
12	.20	.33
1	.20	.26
2	.20	.18
3	.20	.11
4 AM	.20	.03
5	.20	.04
6	.50	.01
7	.90	.11
8	.30	.18
9	.30	.21
10	.30	.22
11	.50	.18
12	.80	.24
1	.50	.16
2	.30	.13
3	.30	.16
4. PM	.50	.24
5	.70	.20
6	.80	.26
7	.80	.30
8	.80	.29
9	.80	.16
10	.50	.26
11	.20	.37
12	.20	.33

TABLE B.12

INFILTRATION AIR FLOW COEFFICIENTS

(from Table A.17, Ref. 2)

Note: These coefficients are used to determine the infiltration air flow rates through:

$$I = C\Delta P^{N}$$

where I = infiltration, CFM per linear crack foot (or per square foot of wall area)**
ΔP = pressure difference across opening, in inches of water

		C	N
1.	Double-hung windows (locked)*		
	non-weatherstripped, loose fit	6	0.66
	average fit	2	0.66
	weatherstripped, loose fit	2	0.66
	average fit	1	0.66
2.	Window frames*		
	masonry frame with no caulking	1.2	0.66
	masonry frame with caulking	0.2	0.66
	wooden frame	1.0	0.66
3.	Swinging doors*		
	1/2" crack	160	0.5
	1/4" crack	80	0.5
	1/8" crack	40	0.5
4.	Walls**		
	8" plain brick	[illegible]	0.8
	8" brick and plaster	0.01	0.8
	13" plain brick	0.8	0.8
	13" brick and plaster	0.004	0.7
	13" brick, furring, lath and plaster	0.03	0.9
	frame wall, lath and plaster	0.01	0.55
	24" shingles on 1x6 boards on 14" centers	9	0.66
	16" shingles on 1x4 boards on 5" centers	5	0.66
	24" shingles on shiplap	3.6	0.7
	16" shingles on shiplap	1.2	0.66

*Values of C listed for these openings are per ft. of linear crack length.

**Values of C listed for the walls are per unit area of the wall surface.

APPENDIX C

HEAT EXCHANGER DESIGN

A wide variety of heat exchanger sizes and designs are commercially available to achieve a given desired heat transfer rating. The ultimate heat exchanger choice for a given application will depend upon the size and design of the associated piping system and upon energy consumption and capital cost criteria established by the designer. Tube fins and vanes and multi-pass flow geometries are commonly used to increase effectively the heat transfer area of a unit without greatly altering its physical dimensions. Tradeoffs among the number of tubes, their lengths and their diameters also affect the total heat transfer area and the fluid pressure losses in the exchanger for a given fluid mass flowrate. Because of its design simplicity and ease of analysis in calculating flowstream and energy transfer effects, a single-pass, counterflow, straight tube heat exchanger geometry was assumed to be used in all of the utility system simulations. The final system design and optimization criteria will not be significantly affected by the actual heat exchanger geometries chosen as long as the specified heat transfer ratings are met and the pressure losses through the units do not greatly exceed those calculated for the straight tube models used in sizing the distribution loop piping and the circulation pumps.

The Tubular Exchanger Manufacturers Association (TEMA) has established a set of design and construction standards to be met by commercially available heat exchangers. [1] Table C.1 lists their preferred tube gages for *Class C* heat exchangers designed for commercial and general process applications. The standard tube bundle patterns are triangular and square matrices, and pressure ratings vary from 150 to 2500 psig. [1] All heat exchangers for the proposed thermal utility system were assumed to have 1" O.D., 14 gage steel tubes in square bundles with 1" diameter interstitial flow channels. The tube diameters were chosen to provide "reasonable" physical dimensions for the heat exchangers within the geometrical constraints of the computer models, but no additional optimization of the exchanger sizes was performed. The same tubes were assumed to be used in each unit to provide uniform design criteria for all the heat exchangers. The heat transfer coefficient for a tube in one of these heat exchangers can be calculated through:

$$U = \frac{1}{\frac{1}{h_o} + r_o + r_w + r_i\left(\frac{A_o}{A_i}\right) + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right)} \qquad \text{(C.1)}$$

where U = heat transfer coefficient in BTU/hrft^2°F referred to the tube outer surface,

h_o = film coefficient of fluid outside tube,

h_i = film coefficient of fluid inside tube,

r_o = fouling resistance of outer surface of tube,

r_i = fouling resistance of inner surface of tube,

r_w = resistance of tube wall referred to outer surface,

A_o = tube outer surface area, and

A_i = tube inner surface area.

According to El-Wakil [2], the film heat transfer coefficient for water can be approximated by:

$$h = 0.00134(T + 100)\,\frac{V^{0.8}}{D_c^{0.2}} \qquad \text{(C.2)}$$

where h = film heat transfer coefficient in BTU/hrft^2°F,

T = bulk fluid temperature or mean film temperature if temperature drop across film is $\geq$ 10 °F,

V = fluid velocity in ft/hr, and

D_c = channel diameter in ft.

The thermal resistance of the tube wall referred to its outer surface is given by TEMA as:

$$r_w = \frac{t_w}{12k_w}\left(\frac{d}{d - t_w}\right) \qquad \text{(C.3)}$$

where r_w = wall thermal resistance in $(BTU/hrft^{2}{}^{\circ}F)^{-1}$,

t_w = tube wall thickness in inches,

k_w = tube wall thermal conductivity in BTU/hrft°F, and

d = tube O.D. in inches.

Finally, the fouling resistance measured for a wide variety of water types and flow velocities is approximately 0.002 $(BTU/hrft^{2}{}^{\circ}F)^{-1}$ for water temperatures above 125 °F. [1]

If an average water temperature of 300 °F inside the tubes, a water temperature of 200 °F in the channels and fluid velocities of 3 feet/second are assumed, the heat transfer coefficient between a 1" O.D., 14 gage carbon steel tube and a 1" I.D. interstitial channel is easily calculated.

1. Tube outer surface film coefficient:

$$h = 0.00134(200 + 100)\left[\frac{(3 \times 3600)^{0.8}}{(\frac{1}{12})^{0.2}}\right]$$

$$= 1113.78 \text{ BTU/hrft}^{2}{}^{\circ}\text{F}$$

2. Tube inner surface film coefficient:

$$h = 0.00134(300 + 100)\left[\frac{(3 \times 3600)^{0.8}}{(\frac{.834}{12})^{0.2}}\right]$$

$$= 1539.95 \text{ BTU/hrft}^{2}{}^{\circ}\text{F}$$

3. Tube wall thermal resistance:

(thermal conductivity of carbon steel = 29 BTU/hrft°F)

$$r_w = \frac{.083}{(12)(29)}\left[\frac{1}{1 - .083}\right]$$

$$= 0.00026\ (\text{BTU/hrft}^2{}^\circ\text{F})^{-1}$$

4. Outer/inner surface area ratio:

$$\frac{A_o}{A_i} = \frac{\pi d_o L}{\pi d_i L} = \frac{d_o}{d_i} = \frac{1}{.834} = 1.199$$

5. Fouling resistances:

$$r_o = r_i = 0.002\ (\text{BTU/hrft}^2{}^\circ\text{F})^{-1}$$

6. Heat transfer coefficient:

$$U = \frac{1}{\frac{1}{1113.78} + .002 + .00026 + .002(1.199) + \frac{1}{1539.95}(1.199)}$$

$$= 157.87\ \text{BTU/hrft}^2{}^\circ\text{F, and}$$

7. Since the tube outer diameter = 1" = .083':

$$U' = \pi d_o U = 41.33\ \text{BTU/hrft}^\circ\text{F}.$$

TABLE C.1

TEMA PREFERRED TUBE GAGES FOR CLASS C HEAT EXCHANGERS

(from Ref. 1)

Tube O.D., inches	BWG	Wall Thickness, inches	Material
1/4	24	.022	Copper
3/8	22	.028	Copper
1/2	20	.035	Copper
5/8	18	.049	Copper
3/4	16	.065	Copper
	14	.083	Steel
	18	.049	Alloy
1	16	.065	Cooper
	14	.083	Steel
1-1/4	14	.083	Steel

Assuming a design fluid velocity of 3 feet/second and a maximum tube length of 80 feet (four 20-foot bundles in series) for each of the heat exchangers in the 100% Thermal case system, the above analysis, combined with the overall heat transfer ratings and average temperatures listed in Table 4.5 of the text, was used to determine the heat exchanger parameters summarized in Table C.2. As is evident from the fluid velocities calculated for these heat exchangers, the assumption of a velocity value of three feet per second does not hold for the actual units input to the simulation models. However, because the heat exchangers represent a very small contribution to the total loop fluid pressure losses, the total pressure losses remain virtually unchanged when the given velocities are adjusted to their nominal design values.

Tables C.3-C.5 present the heat exchanger tube design parameters calculated for 85%, 75%, and 65% thermal supply utility system options described in Sections 4.2-4.4 of the text.

TABLE C.2

100% THERMAL CASE HEAT EXCHANGER TUBES

Heat Exchanger (see Fig. 4.2)	Heat Transfer Coefficient (BTU/hr°Fx10^4)	Number of Tubes(1)	Fluid Velocity(2) (ft/sec)
1	260.3	765	7.71
2	327.2	981	6.00
3	281.7	866	6.80
4	938.9	2966	1.99
5	160.9	506	0.69
6	160.9	506	0.65
7	97.2	306	0.61
8	97.2	306	0.69
9	103.7	329	0.61
10	214.9	682	0.55
11	214.9	682	0.61
12	124.9	682	0.57
13	242.9	786	0.57
14	141.1	456	0.33
15	141.1	456	0.57
16	141.1	456	0.42
17	242.9	786	0.44
18	202.7	661	0.42
19	629.6	2052	0.41
20	629.6	2052	0.42
21	202.7	661	0.41
22	629.6	2062	0.30

(1) Tubes are 1" O.D., 14 gage, 80 ft. long.

(2) Calculated from design loop fluid mass flowrates.

TABLE C.3

85% THERMAL CASE HEAT EXCHANGER TUBES

Heat Exchanger (see Fig. 4.3)	Heat Transfer Coefficient (BTU/hr°Fx10^4)	Number of Tubes(1)	Fluid Velocity(2) (ft/sec)
1	374.1	1101	4.53
2	388.2	1179	4.23
4	1085.0	3436	1.45
5	132.8	417	0.90
6	132.8	417	0.85
7	79.2	249	0.80
8	79.2	249	0.90
13	132.8	417	0.70
9	108.5	346	0.72
10	224.9	718	0.64
11	224.9	718	0.71
12	224.9	718	0.69
18	202.7	661	0.42
19	654.5	2133	0.42
20	654.5	2133	0.41
21	202.7	661	0.41
22	654.5	2133	0.29

(1) Tubes are 1" O.D., 14 gage, 80 ft. long.

(2) Calculated from design loop fluid mass flowrates.

TABLE C.4

75% THERMAL CASE HEAT EXCHANGER TUBES

Heat Exchanger (see Fig. 4.4)	Heat Transfer Coefficient (BTU/hr°Fx10^4)	Number of Tubes(1)	Fluid Velocity(2) (ft/sec)
1	249.4	731	5.47
2	392.7	1188	3.36
4	1295.0	4121	0.97
5	127.6	400	0.91
6	127.6	400	0.85
13	127.6	400	0.72
9	91.9	292	0.78
10	190.4	605	0.70
11	190.4	605	0.78
12	190.4	605	0.73
18	202.7	661	0.42
19	629.6	2052	0.41
20	629.6	2052	0.42
21	202.7	661	0.41
22	629.6	2052	0.30

(1) Tubes are 1" O.D., 14 gage, 80 ft. long.

(2) Calculated from design loop fluid mass flowrates.

TABLE C.5

65% THERMAL CASE HEAT EXCHANGER TUBES

Heat Exchanger (see Fig. 4.5)	Heat Transfer Coefficient (BTU/hr°Fx10^4)	Number of Tubes(1)	Fluid Velocity(2) (ft/sec)
2	471.5	1394	2.54
4	933.0	2944	1.20
5	132.8	417	0.90
9	61.6	193	0.90
10	132.8	417	0.78
11	132.8	417	0.86
12	132.8	417	0.83
18	160.6	522	0.49
19	518.7	1686	0.49
20	518.7	1686	0.48
21	160.6	522	0.48
22	518.7	1686	0.34

(1) Tubes are 1" O.D., 14 gage, 80 ft. long.

(2) Calculated from design loop fluid mass flowrate.

References

1. "Mechanical Standards of TEMA Class 'C' Heat Exchangers," Chapter 6, Standards of the Tubular Exchanger Manufacturers Association, Fifth Edition, Tubular Exchanger Manufacturers Association, New York, 1968.

2. El-Wakil, M., Nuclear Heat Transport, International Textbook Compnay, Scranton, Penn., 1971, p. 243.

APPENDIX D.1

CALCULATION OF HTGR CAPITAL COST

Metcalfe, et al., [1] calculates the costs of HTGR power plants using the CONCEPT III Code [1], arriving at the results shown in Fig. D.1 reproduced from his report [1]. This figure shows the variation of capital cost (in terms of dollars per Kwhr) versus power plant electrical capacity. These data are well-represented by the equation

$$\text{Unit Capacity Cost} = 16650.(\text{Mw(e)})^{-.497} \qquad \text{D.1}$$

where Unit Capacity Cost is stated in terms of 1985 dollars per KW(e), and the quantity - Mw(e) - refers to plant electrical capacity stated in Mw(e).

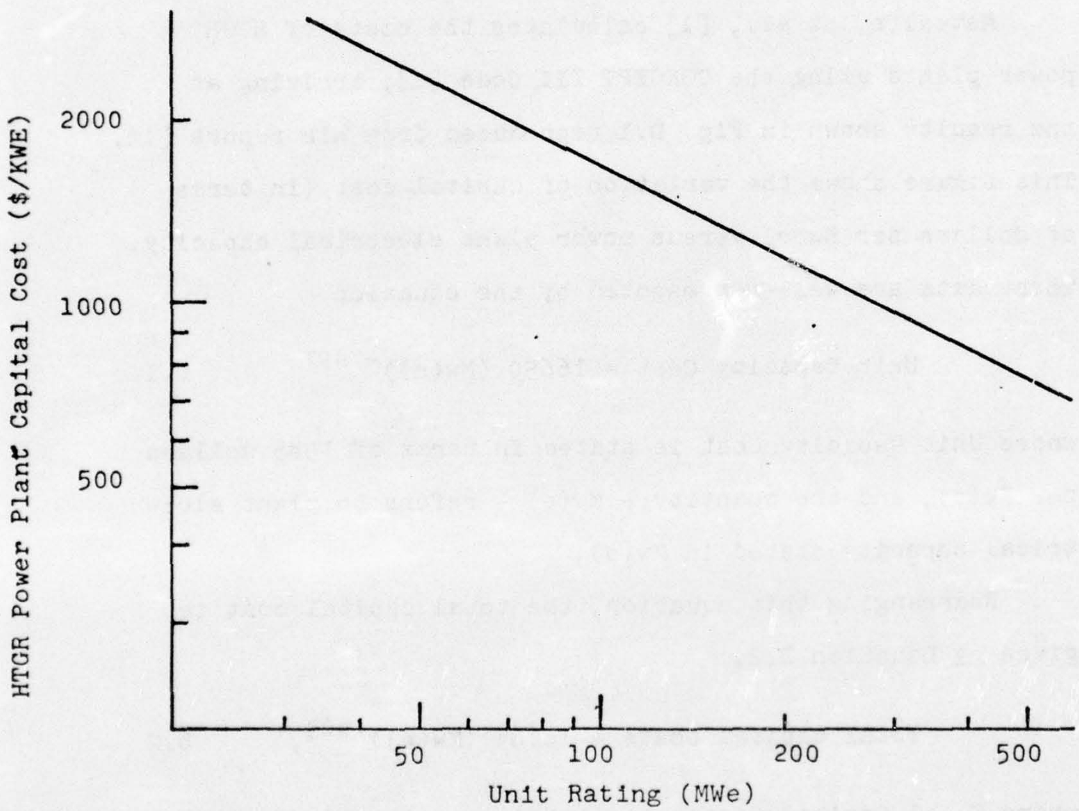
Rearranging this equation, the total capital cost is given by Equation D.2,

$$\text{Total Capital Costs} = 16.65\ (\text{Mw(e)})^{.503}, \qquad \text{D.2}$$

where Total Capital Costs are stated in units of millions of 1985 dollars.

The capital cost calculations are based on the assumptions listed in Appendix D.4.

Fuel costs are calculated using 17.4 mills/KWhr. Total operation and maintenance costs are based on a cost of 7.9 mills/KWhr, both values are taken from Metcalfe's work [1].



Note: All costs are given in 1985 dollars.

Figure D.1. Unit Capital Cost as a Function of Rating

APPENDIX D.2

CALCULATION OF COAL CONSUMPTION

Annual coal consumption for a given thermal/electric split is calculated by integrating the daily average coal consumption rate over the year. Coal consumption for a given day is found from the heat rate for the gas turbine generators and the electric and thermal loads calculated by TDIST (see Chapters 3, 4 and 5). Specifically the sequence of calculations is the following:

1. A twenty-four hour simulation of the Ft. Bragg thermal and electrical power demands for a particular day at a given thermal electric load split is performed (Figure 5.34),
2. The gas consumption required for generation of the electrical energy demanded is calculated by using an average gas turbine heat rate of 13,000 BTU/KW(e)-hr [2],
3. The waste heat recovered from the turbine exhaust (as reported in the project Gas Turbine report [2]) is subtracted from the total thermal energy demand for the day. If the total thermal energy demanded exceeds the waste heat recovered from the turbine exhaust, additional gas is burned in a central hot water heater. (The extra gas which is burned in this fashion is assumed to supply energy at a rate of 5,000 BTU/KW(t)-hr),

AD-A031 744 MASSACHUSETTS INST OF TECH CAMBRIDGE DEPT OF NUCLEAR--ETC F/G 13/1
DESIGN OF A NUCLEAR POWERED TOTAL ENERGY SYSTEM FOR FORT BRAGG,--ETC(U)
MAY 76 J W STETKAR, F R BEST, M W GOLAY DAAK02-74-C-0308

UNCLASSIFIED USAFESA-RT-2011 NL

4 OF 4
AD
A031744

END
DATE
FILMED
12-76

4. The total amount of coal consumed during the day is found by adding gas consumption for electrical energy generation to any extra gas consumption for direct thermal heating and by converting gas consumption to coal consumption via an average gasifier coal-to-gas conversion efficiency of 80%,

5. The yearly coal consumption for a given thermal/electric load split is found by repeating steps 1 through 4 over the desired range of annual weather variation. This provides the basic data for the annual fuel consumption integration. In practice simulations for an average winter day, an average winter-spring day, an average spring-summer day and an average summer day are used in constructing an annual fuel consumption schedule (see Fig. 6.3). The annual fuel consumption data are then integrated over the year to obtain an estimate of the total annual fuel consumption rate.

Steps 1 through5 must be repeated for each thermal/electric load split of interest. Additionally, the winter peak and summer peak design day simulations must be performed, since these days determine the TES maximum loads and load variations, and hence the required power generation and thermal reservoir equipment capacities. Samples of these calculations are the following:

1. Typical TDIST results for an average winter day, 85% thermal/electrical load split are

   Average Thermal Demand 76.1 MW(t), and

   Average Electric Demand 30.2 MW(e);

2. Using a gas turbine heat rate of 13,000 BTU/KW-hr, the day's electrical generation gas consumption would be given as

   $$\text{Gas Consumption for Electrical Generation} = 30.2\text{x}10^3 \text{ KW(t) x 24 hrs. x } 13\text{x}10^3 \text{ BTU/KW-hr, or}$$

   $$\text{Gas Consumption for Electrical Generation} = 9.42\text{x}10^9 \text{ BTU of gas;}$$

3. The waste heat recovered from the generation of this electrical energy (from Ref. 2) would be determined as

   $\dot{Q}$ waste heat exchanger = 985 MW(t)-hrs,

   thus,

   Integral Thermal Demand = $76.1\text{x}10^3$ KW x 24 hrs, and

   Integral Thermal Demand = 1830 MW(t)-hrs

   $\dot{Q}_{\text{waste heat exchanger}}$ = -985 MW(t)-hrs

   Extra heating gas burn = 845 MW(t)-hrs

   (using a heat rate of 5,000 BTU/KW-hr), the extra heating gas burn requires production of

   884 MW(t)-hrs x 5,000 BTU/KW-hr = $4.22\text{x}10^9$ BTU;

4. The total gas consumed for the day is the sum of electrical and heating gas consumption.

$$\text{Total Gas Consumption} = \begin{array}{c}\text{Electrical and Heating} \\ \text{Gas Consumption}\end{array}$$

$$\text{Total Gas Consumption} = 9.42 \times 10^9 \text{BTU} + 4.22 \times 10^9 \text{BTU}, \text{ or}$$

$$\text{TGC} = 13.64 \times 10^9 \text{ BTU}.$$

For a typical gasifier efficiency of 80%, this requires a coal consumption given as

$$\text{Coal Consumption} = \frac{13.64 \times 10^9 \text{ BTU}}{.8}, \text{ or}$$

$$\text{Coal Consumption} = 1.71 \times 10^{10} \text{ BTU};$$

5. Steps 1 through 4 are repeated for the other days of interest for the 85% thermal split TUS (and other splits of interest). From these data Fig. 6.3 is constructed. Integrating the coal consumption rates shown in Fig. 6.3 gives total annual consumption rate versus split as is shown in Fig. 6.4.

REFERENCES

1. Metcalfe, L.J., Driscoll, M.J., "Economic Assessment of Nuclear and Fossil-Fired Energy Systems for DOD Installations," Project Report, Contract No, DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, February 1975.

2. Boyd, W.C., Golay, M.W., "Economic and Technical Aspects of Coal Gasification for Use in Gas Turbine Operation," Project Report, Contract No, DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, 1976.

3. Kelly, J., Golay, M.W., "Economic and Technical Aspects of Gas Turbine Power Stations in Total Energy Applications," Project Report, Contract No. DAAK02-74-C-0308, Department of Nuclear Engineering, MIT, 1976.

APPENDIX D.3

The ultimate and proximate analyses of the coal which is assumed in the study to be consumed is summarized in Table D.3.1.

TABLE D.3.1

ASSUMED COAL ANALYSES

Ultimate Analysis

Carbon	57.1%
Hydrogen	3.9%
Oxygen	8.3%
Nitrogen	.8%
Sulfur	4.5%

Proximate Analysis

Moisture	12.3%
Ash	13.3%
Heating Value	9,500 BTU/lb

APPENDIX D.4

ECONOMIC GROUNDRULES

The economic groundrules used in Estimating TES Costs Over-life are summarized in Table D.4.1.

TABLE D.4.1

ECONOMIC GROUNDRULES USED IN ESTIMATING TES COSTS OVER-LIFE

Plant Types - HTGR/Brayton cycle
CGGT direct cycle

Date of Operation - 1985

Cost of Money - 10%

Average escalation rate - 6.3%

30 year plant lifetime

Straight line debenture accounting

APPENDIX D.5

EQUIVALENT COST OF NATURAL GAS

An example of the calculation of the equivalent breakeven cost of an alternative fuel is presented in the following example:

Case - Coal Costs = \$27/Ton (in 1985)

Thermal/Electrical Load Split = 75%

Cost Breakeven · Mass Annual Fuel = Annual Cost (Capital, Operational Maintenance, and Coal) to run the TES, or

$$\text{Fuel Cost} \times (9.43 \times 3.56\text{x}10^{12}\ \text{BTU}) = \$85.1\text{x}10^{6}$$

→ Breakeven Fuel Cost = \$2.53 per million BTU

APPENDIX D.6

PIPE AND TRENCH COST DATA

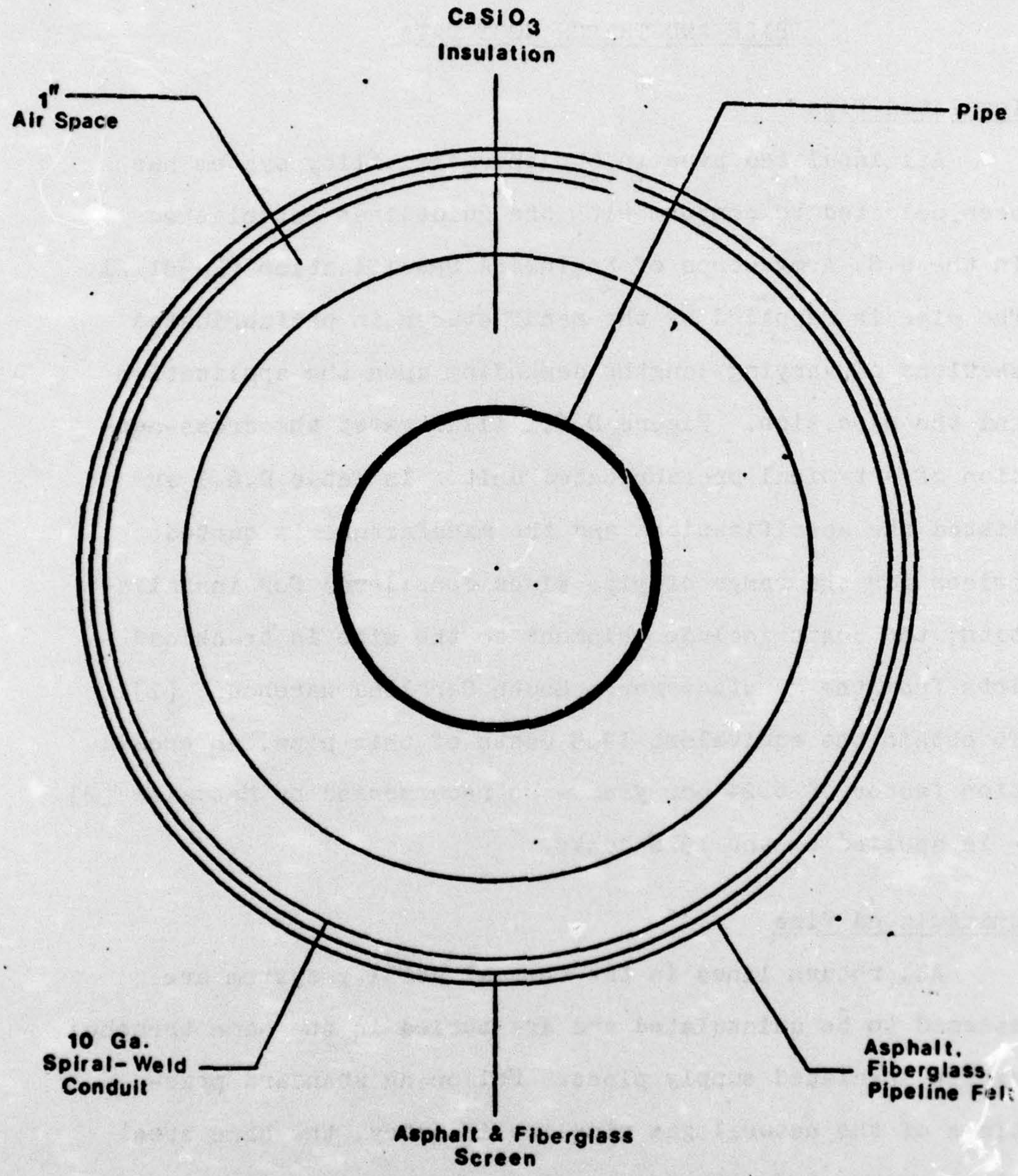
Insulated Pipe

All insulated pipe in the thermal utility system has been selected to conform with the guidelines established in the U.S. Army Corps of Engineers Specification CE-301.21. The pipe is supplied by the manufacturer in prefabricated sections of varying lengths depending upon the application and the pipe size. Figure D.6.1 illustrates the cross-section of a typical prefabricated unit. In Table D.6.1 are listed the specifications and the manufacturer's quoted prices for the range of pipe sizes considered for installation; the costs include shipment to the site in truckload lots from the manufacturer's South Carolina warehouse [1]. To obtain the equivalent 1985 costs of this pipe, an escalation factor of 6.2% per year - as recommended by Metcalfe [2] - is applied to the 1976 costs.

Uninsulated Pipe

All return lines in the thermal utility system are assumed to be uninsulated and are buried in the same trenches as the insulated supply pipes. Following standard practices of the natural gas pipeline industry, the bare steel pipes are coated with a corrosion-preventative compound

Figure D.6.1
Cross-Section of Prefabricated HTW Transmission Pipe



(from Ref. 1)

TABLE D.6.1

PREFABRICATED INSULATED PIPE COSTS

Pipe O.D. (inches)	Pipe Wall Specification	Insulation Thickness (inches)	Jacket O.D. (inches)	1976 Cost(1) ($/ft)	1985 Cost(2) ($/ft)
2	Sched. 40	1-1/2	8-5/8	18	31
3	Sched. 40	2	10-3/4	25	43
4	Sched. 40	2	10-3/4	26	45
6	Sched. 40	2-1/2	14	38	65
8	Sched. 40	2-1/2	16	46	79
10	Sched. 40	2-1/2	19	58	100
12	.375 wall	3	22	81	139
18	.375 wall	4	30	132	227
24	.375 wall	4	36	172	296

(1) From Ref. 1, includes shipping to site in truckload lots.

(2) Escalated at 6.2% per year from 1976.

and are wrapped with protective material for direct soil burial. Table D.6.2 summarizes the cost data used for this pipe. For sizes of 8" O.D. and less, the 1976 costs have been obtained from a local Boston supply company [3], quoted for truckload lots. Costs for the larger pipes are unavailable from local distributors and have been scaled from national building cost file data [4] according to a cost function derived from the small pipe quotations. The 1985 costs are obtained by escalating the 1976 costs at 6.2% per year.

Trenches

The trench cost data summarized in Table D.6.3 are based upon unit costs obtained from an eastern regional construction cost file [5]. Trench dimensions correspond to the HTW pipe manufacturer's specifications for double-circuit burial at a centerline depth of 6 feet [1]. Excavation is assumed to be conducted in average damp sandy loam soil with the use of a trenching machine or backhoe. Backfilling is by bulldozer or backhoe from fill deposited at the trench edge, and the backfilled soil is compacted with an air-powered tamping machine. 1985 costs are obtained by escalating the 1976 data at 6.2% per year.

TABLE D.6.2

COATED UNINSULATED PIPE COSTS

Pipe O.D. (inches)	Pipe Wall Specification	1976 Costs[1] ($/ft)	1985 Costs[2] ($/ft)
2	Sched. 40	1.74	2.99
3	Sched. 40	3.29	5.65
4	Sched. 40	4.74	8.15
6	Sched. 40	8.90	15.29
8	Sched. 40	12.99	22.32
10	Sched. 40	20.87	35.86
12	.375 wall	31.74	54.54
18	.375 wall	49.59	85.22
24	.375 wall	70.99	121.99

(1) Costs for sizes 2"-8" obtained from Ref. 3; costs for sizes 10"-24" scaled from Ref. 4.

(2) 1976 costs escalated at 6.2% per year.

TABLE D.6.3

TRENCHING COSTS

Pipe O.D. (inches)	Trench Dimensions (feet)	Excavation Cost(1) ($/lin.ft.)	Backfill Cost(2) ($/lin.ft.)	1976 Total Cost ($/lin.ft.)	1985 Total Cost ($/lin.ft.)
2	4 x 6-1/2	1.54	2.94	4.48	7.69
3	4 x 6-1/2	1.54	2.94	4.48	7.69
4	4 x 6-1/2	1.54	2.94	4.48	7.69
6	4 x 6-1/2	1.54	2.94	4.48	7.69
8	4 x 7	1.66	3.16	4.82	8.29
10	4-1/2 x 7	1.87	3.56	5.43	9.33
12	5 x 7	2.07	3.95	6.02	10.35
18	6-1/2 x 7-1/2	2.89	5.51	8.40	14.43
24	7-1/2 x 7-1/2	3.33	6.35	9.68	16.64

(1) Base cost = $1.60/cubic yard [5].

(2) Base cost = $3.05/cubic yard [5].

(3) 1976 costs escalated at 6.2% per year.

REFERENCES

1. Personal communication, representative of Kenyon-Barstow Co., Division of Ric-Wil, Inc.

2. Metcalfe, L.J., "Economic Assessment of Alternative Total Energy Systems for Large Military Installations," MIT Department of Nuclear Engineering, S.M. Thesis, August 1976.

3. Personal communication, sales representative of O'Connell Supply Co., Everett, Mass.

4. Building Cost File, 1976 Unit Prices, Eastern Edition, Construction Publishing Co., Inc., New York, 1976.

5. Building Construction Cost Data 1976, Robert S. Means Co., Inc., Duxbury, Mass., 1976.

APPENDIX D.7

PUMPING POWER COSTS AND PUMP RATING CALCULATIONS

Pumping Power

The pumping power required to overcome a given fluid frictional pressure loss is given by Eq.(D.7.1).

$$W = \frac{\Delta P A_c V}{737.56} = \frac{\Delta P \dot{m}}{737.56\rho} \qquad \text{(D.7.1)}$$

where W = pumping power (kW),

ΔP = fluid pressure drop (lbf/ft^2),

A_c = flow channel cross section (ft^2),

V = fluid velocity (ft/sec),

$\dot{m}$ = fluid mass flowrate (lbm/sec),

ρ = fluid density (lbm/ft^3), and

1kW = 737.56 ft-lbf/sec.

Thus, knowing the annual-average fluid mass flowrate and the pipe dimensions for each loop, the Darcy pressure drop formula (Eq.(D.7.2)) may be used to compute the average fluid frictional pressure losses, which are used in Eq.(D.7.1) to determine the average pumping power requirements for the loop.

$$\Delta P = f \frac{L}{D} \frac{\rho V^2}{2g_c} \qquad \text{(D.7.2)}$$

where ΔP = fluid pressure drop (lbf/ft^2),

L = flow channel length (ft),

D = flow channel diameter (ft),

ρ = fluid density (lbm/ft^3),

V = fluid velocity (ft/hr),

g_c = conversion factor = 4.17×10^8 $lbm\text{-}ft/lbf\text{-}hr^2$,

f = Darch-Weisbach friction factor, and

$$f \approx \frac{0.184}{Re^{0.2}} ,$$

where Re is the fluid Reynolds number for turbulent flow.

<u>Pumping Power Costs</u>

If the yearly-average fluid flowrates are used in Eqs.(D.7.1) and (D.7.2) to determine the average pumping power required for each loop, the total pumping power costs over the 30-year lifetime of the thermal utility system may be calculated from Eq.(D.7.3).

$$C_p = 8766 \; W \; N \; C_e \qquad \text{(D.7.3)}$$

where C_p = pumping power costs over the system life($),

W = annual-average pumping power (kW),

N = operational life of the utility system (yr),

C_e = levelized cost of electricity over the system life ($/kWhr), and

1 yr = 8766 hr.

The levelized cost of electricity used in this calculation should, of course, be the time-averaged cost of the electricity produced by the TES power plant, and it will depend strongly upon such factors as the type of plant used, the average thermal/electrical energy demand ratio, fuel costs, the method of system financing, variations in the market interest rate over the 30-year period, and the method chosen for allocating the TES life-costs between its two energy products. Since many of these variables depend critically upon the financial structure of the TES sponsoring authority and upon many non-quantifiable public service and equity considerations beyond the scope of this analysis, no attempt is made to fix upon any single electricity cost as being the optimal value for a particular system configuration. However, in order to translate the pump energy consumption data into representative lifetime costs, a value of 58 mills/kWhr is assigned to the levelized electricity cost - based upon the use of a nuclear HTGR power plant to supply the 75% thermal/electrical load split utility system option - charging aginst the electrical energy generated all capital and fuel costs except those directly associated with the thermal utility system and its unique supply equipment at the central power station. As is shown in Fig. 6.8 of the text, the total system pumping power costs calculated from this electricity cost are very small compared with the costs of the other major TES components, and a substantial variation in the cost of electricity will have negligible effects - from the standpoint of the pumping power costs - upon the overall system choice.

Pump Rating and Costs

Although the average utility system fluid flowrates are determined primarily by the thermal energy demands experienced during the spring and fall months, the pumps must be sized to supply the peak system design conditions, and they operate at relatively low capacity factors throughout most of the year. Equation (D.7.4) can be used to convert the design fluid mass flowrates from units of pounds per hour to units of gallons per minute, which can be used directly in the centrifugal pump cost function shown in Fig. D.7.1, adapted from the work of Ayorinde [1].

$$GPM = \frac{7.48\dot{m}}{60\rho} \qquad (D.7.4)$$

where GPM = fluid volume flowrate (gal/min),

$\dot{m}$ = fluid mass flowrate (lbm/hr),

ρ = fluid density (lbm/ft^3),

1 hr = 60 min, and

1 ft^3 = 7.48 gal.

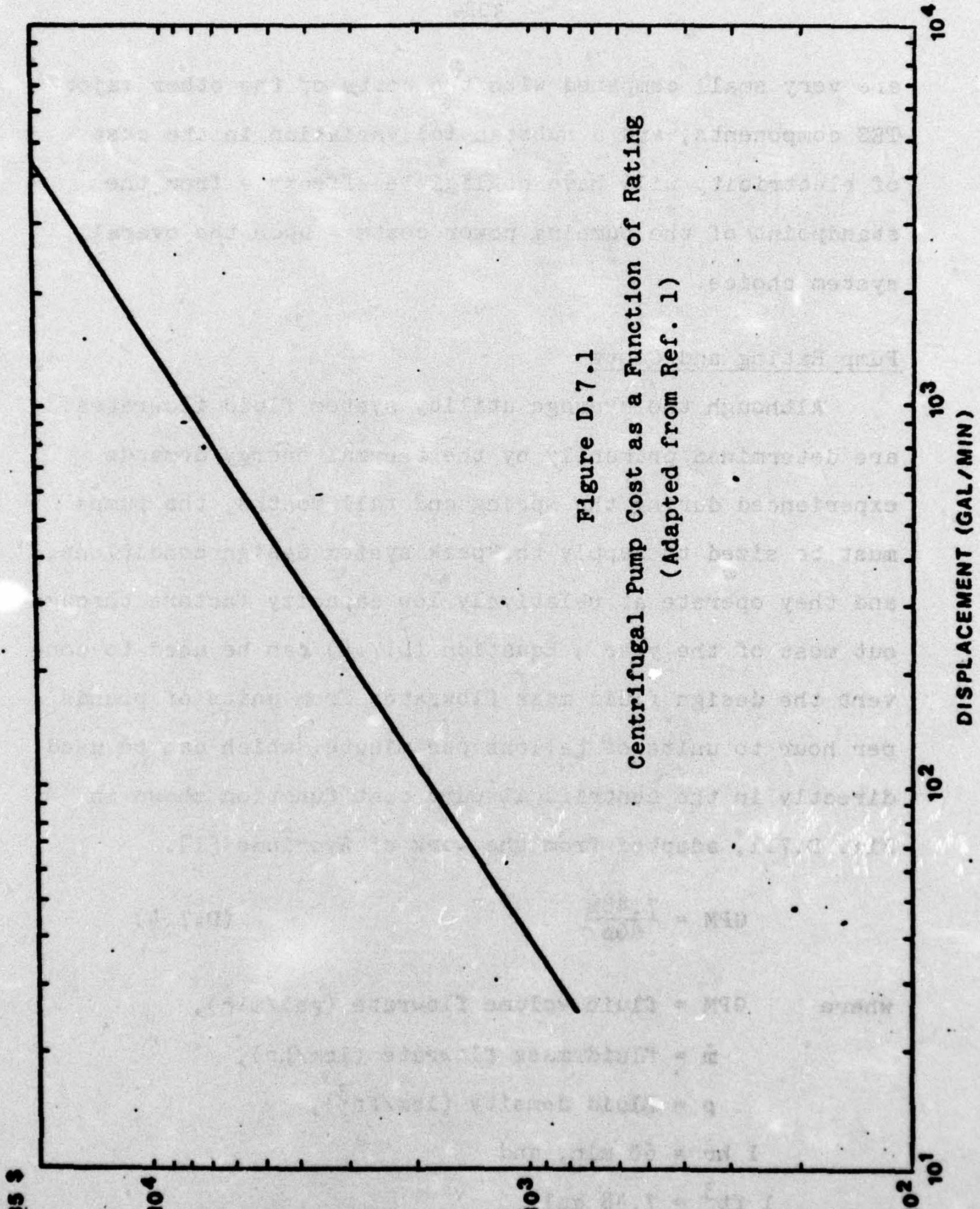


Figure D.7.1
Centrifugal Pump Cost as a Function of Rating
(Adapted from Ref. 1)

Although the costs in Ayorinde's work are presented in 1973 dollars, the cost function shown in Fig. D.7.1 has been escalated at 6.2% per year - following the work of Metcalfe [2] - to obtain equivalent 1985 pump costs. Due to excessive pump component loading, the maximum pump rating recommended for general applications is 3000-3500 gpm [1]. In cases requiring ratings larger than this limit, it is assumed that two or more units are installed to divide the load equally.

REFERENCES

1. Ayorinde, E.O., "Underground Transmission of Heat," MIT Department of Mechanical Engineering, S.M. Thesis, August 1973.

2. Metcalfe, L.J., "Economic Assessment of Alternative Total Energy Systems for Large Military Installations," MIT Department of Nuclear Engineering, S.M. Thesis, August 1975.