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ACRONYMS

ASHRAE	American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc.
ASME	American Society of Mechanical Engineers
CHCP	Central Heating and Cooling Plant (Building 24)
CLCWA	Clear Lake City Water Authority
COP	coefficient of performance
DWH	domestic hot water
ESOP	Energy Systems Optimization Program (computer)
GSA/PBS	General Services Administration/Public Building Service
HVAC	heating, ventilation, and air-conditioning
IUS	integrated utility system
JSC	Lyndon B. Johnson Space Center
MIST	MIUS integration and subsystems test
MIUS	modular integrated utility system
MYE	man-year equivalent
O&M	operations and maintenance
PER	preliminary engineering report
URDC	Urban Research and Development Corporation
USPO	Urban Systems Project Office

STUDY OF LYNDON B. JOHNSON SPACE CENTER

UTILITY SYSTEMS

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SUMMARY

Many energy-saving utility concepts were evaluated during the NASA Lyndon B. Johnson Space Center integrated utility system study conducted in 1975 and 1976. Most of the concepts were derived from energy-saving proposals that either had previously received very little in-depth analysis or were evaluated at a time when energy economics were considerably different from those at the time of this study. The variety of subjects addressed, ranging from fuel cost and availability to architectural innovations for achieving building load reductions, required a multidisciplinary team effort consisting of both in-house and contractor personnel.

The objective of the study was to define and analyze utility options that would provide facility energy savings in addition to the approximately 25 percent already achieved through an energy loads reduction program. A systems engineering approach was used to determine total system energy and cost savings resulting from each of the 10 major options investigated. The study indicated that annual energy savings of as much as 46 percent of the consumption in the 1974 baseline year could be obtained with integrated utility system concepts.

A major recommendation from the study was to convert at least three of the seven existing steam-turbine-driven centrifugal chillers to electric motor drive. The resulting energy saving would be 22 percent of the baseline year consumption, and the investment payback time would be approximately 1.4 years, based on projected fuel and electricity prices. It was also recommended that the following options be periodically reviewed with respect to the cost of fuel: (1) engine-generator/electric-motor-driven centrifugal chiller, (2) solid-waste incineration with heat recovery, and (3) energy storage (chilled water) as applied to the electric-drive chiller option. Other recommendations include implementation of boiler stack gas heat recovery and implementation of a program to reduce pretreated outside ventilation air in selected buildings and thereby to achieve further reduction of heating and cooling loads.

Table I summarizes the results obtained from applying the various energy-saving concepts to the NASA Lyndon B. Johnson Space Center utility system. The energy savings are shown as a percentage of the baseline 1974 total facility energy consumption and also in terms of energy form (electricity or fuel). Also shown are the capital costs and simple payback periods for each option based on projected energy prices. The system configurations and energy-saving options are defined in the section entitled "Design Concepts and Analysis."

INTRODUCTION

The purpose of this report is to document the results of the NASA Lyndon B. Johnson Space Center (JSC) integrated utility system (IUS) in-house study. This study was aimed toward definition and analysis of potential energy-saving utility system modifications for the JSC. The study was initiated in mid-January 1975 as a result of JSC management's interest in pursuing facility energy savings in addition to the approximately 25 percent already achieved through the energy loads reduction program.

The objective of the study effort was to define, analyze, and document potential energy-saving utility system modifications that would serve as a basis for a statement of work to obtain a preliminary engineering report (PER) for the most desirable design concepts. Similar studies have been previously conducted at JSC; however, they either addressed only specific item modifications or they were conducted under entirely different guidelines and constraints. For example, in the original PER for the JSC utility heating and cooling plant,[1] many of the options addressed in this study were considered; however, in 1961 when the study was conducted, the cost of natural gas was a factor of 10 lower than in 1976, and the cost of electricity was a factor of 2.5 less than in 1976. These costs strongly influence equipment and system design considerations.

The study plan provided for utilization of support contractors and consulting engineering support to augment and complement in-house expertise. Specific consultation support was required in the area of fuel availability and energy cost projections for the Houston area. This work was performed by an engineering company under an existing contract with the Office of Facilities, NASA Headquarters. Other contractor support was used in the following areas.

1. Energy systems computer analysis for the Urban Systems Project Office (USPO)

2. Cost and economic analysis for the USPO

3. Specific engineering study tasks

4. Architectural support to the USPO

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[1]Central Plant Facilities for the Manned Spacecraft Center, NASA, Clear Lake City, Harris County, Texas. Prepared by Bernard Johnson and Associates for Brown and Root, Inc., December 1961.

This report begins with a discussion of the existing JSC utility system, which was used as the baseline for comparisons with modification options. This discussion is followed by a listing of study guidelines and a detailed description and analysis of the individual options. Appendix A consists of a description and discussion of a computer program developed to analyze energy flow and equipment performance in the JSC Central Heating and Cooling Plant (CHCP). Appendix B is a discussion of results of computer analysis of JSC buildings to determine heating and cooling load as functions of ambient weather conditions, occupancy, operating equipment, etc., and as a result of modifications designed to reduce these loads.

As an aid to the reader, where necessary the original units of measure have been converted to the equivalent value in the Systeme International d'Unites (SI). The SI units are written first, and the original units are written parenthetically thereafter.

The following JSC civil service personnel contributed significantly to this report: Carl A. Romero, Clyde O. Waters, Glenn W. Spencer, Kornel Nagy, and Walter H. Smith of the Engineering Division, Center Operations Directorate, and James O. Rippey, Richard C. Wadle, Vernon E. Shields, Harman L. Roberts, and Steven P. Wallin of the Systems Evaluation Office (formerly Urban Systems Project Office), Engineering and Development Directorate.

BASELINE JSC UTILITY SYSTEM

In this section, the baseline JSC utility system is described and utility loads and energy consumption are discussed.

Description

The utility system functions of primary interest in this study are as follows.

1. Electrical power

2. Environmental control of buildings (space heating and cooling)

3. Hot-water heating

4. Solid-waste management

5. Potable water supply and wastewater management

The existing utility system includes an 862-kN/m^2 (125 psig) compressed air system, which was included only as a facility load. No studies were made of means to improve this utility function.

Electrical energy is supplied to JSC by parallel transmission lines of 138 kilovolts each from two generating stations of the local commercial power supplier. This energy is stepped down to 12.4 kilovolts at the main JSC substation (number 221) for distribution to the Center utility tunnel system and individual buildings. Further stepdowns, based on user requirements, are performed with transformers in the individual buildings.

The CHCP located in and adjacent to building 24 (fig. 1) provides environmental control utilities. Building 24 supplies and controls 862-kN/m^2 (125 psig) steam at 491 K (425° F) from five natural-gas-fired boilers (2930 kN/m^2 (425 psig), 589-K (600° F)) through the utility tunnel and to the JSC mall buildings. The plant also supplies 279-K (42° F) chilled water to the JSC mall buildings. The chilled water is produced by steam-turbine-driven centrifugal chillers using 2930-kN/m^2 (425 psig), 589-K (600° F) steam. Figure 1(b) is a schematic diagram of the major equipment and services of building 24. Adjacent to building 24 is the cooling tower for the CHCP. The cooling tower is capable of supplying 159 m^3/min (42 000 gal/min) of cooling water to the chiller condensers, the surface condensers, and the utility air compressor aftercoolers. These services are of primary interest to the report because they constitute the major energy requirements of the Center.

In the past, potable water for JSC was pumped from onsite wells and treated by Center facilities. However, potable water is now provided by the local water authority from a surface source to minimize subsidence and the effect of low water levels and high-usage problems. Large quantities of potable water are used in boiler feed-water, cooling-tower supply, and irrigation requirements.

The wastewater flowing from the JSC site is treated by a nearby city wastewater treatment plant. Originally, wastewater was treated onsite; but, as the Center expanded and discharge requirements became more stringent, treatment by the larger offsite facility became desirable.

The collection and disposal of solid waste is a contracted service at JSC. The waste is transported offsite for eventual landfill.

Utility Loads and Energy Consumption

The major loads and energy consumption at JSC are shown in this section by major subsystem and are summarized in figure 2.

Electrical power loads.- The daily electrical load profiles for the following are given in table II.

1. Average fall day

2. Average winter day

3. Average spring day

4. Average summer day

5. Average weekend day/holiday

6. Peak summer day

The loads are based on monthly metered electrical consumption for JSC for the calendar year February 1974 through January 1975. (See table III.) The weekend day/holiday profiles are based on average consumption of 11 500 kilowatts for the total weekend days/holidays for the year.

The peak apparent power and the energy consumption for each month are shown in table III. The demand peaks are determined by averaging the four highest peaks during each month. It may be noted that the higher peaks usually occur in the months characterized by lower energy consumption. This occurrence is a result of the utility rate structure, which encourages the use of more electricity during the winter months. For IUS design purposes, the winter season peaks may be managed within the summer demand peaks.

In developing the average daily profiles, the weekend day/holiday electrical energy consumptions were subtracted from the total seasonal consumption and divided by the number of workdays for each season. This computation gives the average workday consumption for the season. The hourly consumption is determined by matching the hourly demand ratio (minimum demand to peak demand) to the average workday profiles measured at JSC for the week of June 1 to 7, 1975. The average daily peaks are the average of the seasonal peaks or 22 200 kilowatts, whichever is less.

Heating and cooling loads.- Primary data sources for establishing JSC heating and cooling loads were building 24 boiler and chiller records. These records show hourly flow rates and hourly instantaneous loads, but do not indicate cumulative or total energy generated or used. These records are made hourly for each of the boilers and chillers in operation. Monthly summary records have been generated, and these records have helped to determine maximum and minimum days as well as representative average seasonal periods.

Representative daily profiles for the winter, summer, and fall seasons were selected by comparing recorded local ambient conditions during 1974 with average seasonal temperature ranges for this area. The days which most nearly approximated the ranges for each representative season for both workdays and weekends/holidays determined the JSC plant data used. In several cases, plant data for those days were either incomplete or missing; therefore, other representative days were selected. For analysis purposes, the fall season daily profile was used for a representative spring day. Figures 3 and 4 illustrate the heating, ventilation, and air-conditioning (HVAC) load profiles and daily totals. Figure 3 shows the 1974 JSC maximum, minimum, and seasonal cooling loads and their representative totals. Initial examination and plotting of the combined space-heating and domestic-hot-water loads calculated indicated no easily distinguishable levels or trends; therefore, only the maximum and minimum daily profiles and the tabulated totals are provided. For the computer program analysis, the maximum heating profile was used for the fall, winter, and spring seasons and the minimum heating profile was used for the summer. Figure 4 contains representative maximum and minimum heating profiles.

The baseline presented in figure 2 was the best information available at the time this study was initiated. Later surveys indicate different proportions of natural gas consumption than indicated in figure 2. Although figure 2 shows 57 989 gigajoules (55×10^9 British thermal units) (4 percent of the total) going to "other," the actual usage in other buildings is closer to 20 percent, or 263 588 gigajoules (250×10^9 British thermal units), during a typical year. However, all options in this study are based on the same original baseline information and therefore can be compared with each other on a relative scale. The effect of the discrepancy would be to slightly reduce energy savings and thereby to decrease total savings and slightly increase payout periods.

Water and wastewater treatment.- The data used in the definition of the loads model for the JSC water systems were obtained from water records and with the use of theoretical calculations on water required for irrigation. The data used were in the form of engineering reports and surveys, water well pumping logs, wastewater treatment bills, metered water usage data, and verbal data based on individual experience. The irrigation water requirements were determined using the sprinkling demand calculated from the formula given in a Johns Hopkins University report entitled "Residential Water Use," knowledge of the area irrigated, and the rationale used in sprinkling at JSC. The potable water usage was determined from the monthly water records for January through December 1974. These are records of the daily quantity of water pumped from the onsite (JSC) wells. The data were summarized into monthly and seasonal usage and then arranged to obtain the average daily flow per season.

The monthly water volume needed for boiler makeup was determined from meters on the boilers in building 24. Additionally, the total cooling-tower-water makeup volume for the year was estimated at 632 164 cubic meters (167×10^6 gallons), based on chiller operation. The corresponding volume of yearly cooling-tower blowdown was 132 489 cubic meters (35×10^6 gallons). Table IV shows the estimated monthly water demands for all major water users and the monthly wastewater loads.

Solid-waste disposal.- The JSC solid waste consists of 3143.4 megagrams (3465 tons) per year of type 0 waste (19.76 MJ/kg (8500 Btu/lb)) collected in refuse containers and 2299.7 megagrams (2535 tons) per year of other wastes of combined types 0, 1, and 2 (13.95 MJ/kg (6000 Btu/lb) estimated). (Types 0, 1, and 2 waste are used as defined by Standards of the Incinerator Institute of America. These data are used for computation involving energy available from the solid waste.) This solid waste is generated during 252 days per year (weekends and holidays are assumed zero). The total JSC solid waste (5443.1 Mg/yr (6000 tons/yr)) is currently collected and disposed of in an offsite landfill by a private contractor.

For purposes of energy analysis, a constant solid-waste production rate of 104.3 megagrams (115 tons) per week with average heating value of 18.47 MJ/kg (7994 Btu/lb) was assumed. This heating value is based on a weighted average of the solid-waste mix indicated in the preceding paragraph. The primary component of the solid waste is paper with very little inorganic or moisture content. Should future data indicate the presence of more inorganics or moisture in the waste system, some adjustment to the heating value will be necessary.

GUIDELINES AND CONSTRAINTS

The guidelines and constraints established for conducting the study were the following.

Design Constraints

The following design constraints were established.

1. Design concepts shall consist of bondable off-the-shelf hardware and equipment: however, in phased-buildup concepts, consideration shall be given to newly developed equipment that has undergone significant testing and evaluation.

2. The basic potable water supply shall be a surface water supply.

3. Consideration may be given to the use of incinerators and/or pyrolysis units for disposal of JSC solid waste and for energy recovery. The implications of importing solid waste shall be addressed. Residue from such units would be hauled away to landfill or other suitable disposal sites.

4. Consideration may be given to the treatment of JSC liquid waste and reuse of water for equipment-cooling makeup water supply and for irrigation and other nonpotable purposes.

5. Consideration may be given to onsite electrical power generation using natural gas, fuel oil, coal, and potential synthetic fuels derived from solid wastes and coal.

6. Maximum utilization shall be made of existing chilled water, compressed air and steam distribution systems, building HVAC systems, and the utility control system.

7. Maximum utilization shall be made of existing electrical power distribution systems.

8. Consideration shall be given to phased installation of new equipment corresponding to planned retirement (or major overhaul) of existing equipment.

9. The improved utility systems shall comply with applicable environmental pollution regulations.

10. The utility systems shall service the utility loads (power, HVAC, water, and solid waste, as applicable) with no degradation in quality (except as a design parameter) and with reliability equal or superior to a "conventional" or existing system.

11. HVAC design and analysis shall comply with prevailing guidelines for comfort and for critical environmental areas.

12. Consideration may be given to the use of thermal storage systems.

Cost Guidelines

The following cost guidelines were adopted.

1. Fuel and electrical energy cost projections shall be as given in the contractor-prepared study report.[2] Table V contains the fuel and electrical power prices for the baseline-case scenario that was used in the study.

2. Escalation rates on all items except fuel and electricity shall be the following.

Time period	Rate
1975 to 1980	10 percent/yr
Post-1980	5 percent/yr

DESIGN CONCEPTS AND ANALYSIS

In the initial design analyses of JSC utilities prior to the construction phase, numerous options were considered. Energy costs in 1961 were much less a consideration than availability of major equipment. Nevertheless, detailed analysis was made of several alternate concepts described in the engineering study and report[1] prepared at that time.

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[1] Central Plant Facilities for the Manned Spacecraft Center, NASA, Clear Lake City, Harris County, Texas. Prepared by Bernard Johnson and Associates for Brown and Root, Inc., December 1961.

[2] Energy Outlook and Alternate Fuels Study - 1985-2000. Prepared by the Ralph M. Parsons Company for the NASA Johnson Space Center (job. no. 5494-1), November 1975.

Since 1961, the energy picture has changed dramatically, necessitating a reevaluation of JSC utilities. The following sections are based on this new perspective and the fact that a major investment in equipment is already on hand and operational. The specific system concepts investigated and studies performed for this report are as follows.

1. Total energy systems

2. Engine-generator/electric-motor-driven chiller systems

3. Electric-motor-driven chillers

4. Combination steam-turbine-driven compression chillers and absorption chillers

5. Solid-waste management with energy recovery

6. JSC water and wastewater treatment considerations

7. Energy storage concepts

8. Solar energy applications

9. Boiler stack heat recovery

10. Alternate fuels evaluation

11. Building/site heating and cooling loads reduction options. (For load reduction options, see appendix B.)

A system description is provided. The description is followed by an energy analysis of baseline and alternate configurations and an economic analysis.

Total Energy Systems

Conventional methods of providing electric utility services use steam- or gas-turbine-driven generators located remotely from the areas they serve. This practice prohibits the economical utilization of vast quantities of heat (typically 65 percent of the fuel heating value) normally rejected in the electrical energy production cycle. In addition, the requirement for transmission lines extending long distances to deliver the electrical energy from the generating site to the user results in further energy losses in the system.

The objectives of the total energy concept are to maximize the use of the energy contained in the fuel and to simultaneously produce electricity efficiently. The heart of the total energy system is the prime mover. Reciprocating engines (gas, diesel, and dual-fuel) and gas turbines are all well adapted to the total energy concept. They are commercially available in a variety of sizes with accessory heat-recovery equipment for obtaining heat energy from exhausts, water jackets, lubrication oil circuits, and intercoolers. Location of total energy systems near the use point allows for an efficient distribution of electrical energy as well as economical distribution and utilization of waste heat.

In the total energy system design presented herein for the JSC facility, primary emphasis was placed on high electrical conversion efficiency as a basis for the hardware selection. The candidate prime-mover systems were boiler/steam turbine, gas turbine, and reciprocating engines. Large, low-speed, dual-fuel, reciprocating engine generator units typically have heat rates in the 2.34- to 2.64-J/J (8000 to 9000 Btu/kWh) range. This fact is illustrated in figure 5, where curves of heat rate compared to percent electrical load for representative steam turbo-generator, gas turbine-generator, and dual-fuel, reciprocating engine-generator systems are shown. By comparison, typical heat rates for large control station powerplants are in the 2.93- to 3.22-J/J (10 000 to 11 000 Btu/kWh) range. It is noted that in the unit power range of interest (5000 to 10 000 kilowatts), the reciprocating engine system has the lowest heat rate (highest efficiency).

Fuel considerations.- Total energy systems utilizing internal combustion (IC) engines as prime movers require either liquid or gaseous fuels, or both simultaneously in the case of dual-fuel IC engines. Steam powerplants may be coal-fired because external combustion boilers are used. Cost projections from a fuels availability study[2] indicate that natural gas cost will exceed fuel oil (no. 2 diesel) cost in 1977 or early 1978. However, it was assumed in cost comparisons that the existing baseline system would continue use of natural gas. Based on these considerations, reciprocating engine prime movers were selected for total energy system conceptual design purposes. A description of the design and its auxiliary equipment is given in the following section.

System description.- The total energy system designed for the JSC facility utilizes six turbocharged, dual-fuel engines rated at 7271 kilowatts (9750 horsepower) each, and producing 7025 kilowatts of 60-hertz power at the generator bus. A diagram of the system is shown in figure 6. These units incorporate heat recovery from the exhaust system in the form of 394.26-K (250° F), 103.4-kN/m^2 (15 psig) saturated steam. This energy is used to operate absorption chillers. Heat produced in the water jackets and the lubrication oil circuits is rejected through an airblast heat exchanger to maintain an engine operating temperature of 349.82 K (170° F).

The generator voltage outputs are tied through switchgear to a common three-phase bus, which is stepped up through a transformer to a 12.4-kilovolt, three-phase system feeding the main distribution bus. A diagram of this interconnection with the main substation (facility number 221) is given in figure 7.

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[2]Energy Outlook and Alternate Fuels Study - 1985-2000. Prepared by the Ralph M. Parsons Company for the NASA Johnson Space Center (job. no. 5494-1), November 1975.

The peak cooling loads are met by two absorption chillers and three compression chillers. The system design includes the addition of three absorption chillers and electric motors to drive fans for the existing compression chillers. Cooling-tower requirements for the total energy option are 131.73 m^3/min (34 800 gal/min), or approximately 83 percent of the existing capacity. The following additional major equipment is required for this option.

1. Six dual-fuel prime-mover/generator sets

    a. Exhaust-heat-recovery boiler and controls

    b. Oil cooler/intercooler heat exchanger and controls (air cooled)

    c. Jacket-water heat exchanger and controls (air cooled)

    d. Holding tanks

    e. Pumps: 480 volts alternating current

2. Six 9000-kilovolt-ampere switchgear units (oil breaker)

3. Automatic paralleling equipment for each prime mover

4. One 30-megavolt-ampere step-up transformer: three phase, 600 volts/12.4 kilovolts

5. Three absorption chillers (4843 kilowatts (1337 tons) each)

6. Four electric motors: 1678.5 kilowatts (2250 horsepower), synchronous

7. One stepdown transformer: 12.4 kilovolts/2400 volts

8. Three 15-kilovolt circuit breakers for circuit to substation

Optional configurations.- The basic total energy system concept previously described may be expanded to include energy input from JSC solid-waste processing. A later section of this report (Solid-Waste Management/Energy Recovery) describes solid-waste incineration and pyrolysis processes applicable to the JSC site. With incineration, the solid-waste energy could be recovered in the form of low-pressure (103.4 kN/m^2 (15 psig)) steam; and using commercially available waste heat boilers, this steam could be piped directly into the 103.4-kN/m^2 (15 psig) header (fig. 6) for producing additional absorption air-conditioning. On the other hand, incinerator boilers could generate 1620-kN/m^2 (235 psig), 491.48-K (425° F) steam for direct use in the medium-pressure steam header and for distribution to JSC buildings. The efficiency of heat recovery is slightly less in the latter case because of higher exhaust gas temperatures, but the steam utilization efficiency is higher. Therefore, the latter configuration was selected for analysis.

Pyrolysis of the solid waste would produce gaseous fuel with a heating value of 5585 to 20 479 kJ/m^3 (150 to 550 Btu/ft^3), depending upon process selection. This fuel gas can be utilized in the dual-fuel engines to generate electrical power. The fuel may be blended with natural gas, if that is the primary fuel, or it may be used as a fuel supplement by introducing the gas into the air inlet of engines that use diesel oil as the primary fuel. The latter process is called fumigation.

Tests have been conducted at the JSC modular integrated utility system (MIUS) integration and subsystems test (MIST) laboratory to demonstrate fumigation utilizing a 298-kilowatt (400 horsepower), six-cylinder, four-cycle, precombustion chamber-type engine. In these tests, as much as 20 percent of the total energy consumed by the diesel engine was gaseous fuel injected into the air inlet. The tests indicated excellent utilization of the gas energy; stable engine operation from 25 percent to 100 percent load; and no predetonation, which would be deleterious to engine life.

Another potential variation of the basic total energy system would be utilization of recovered oil-cooler and jacket-water heat from the dual-fuel engines. To use this heat, however, the JSC distribution system would have to be changed from the existing steam system to a 352.59-K (175° F) hot-water system. This system modification was studied by an engineering company, and the results of the study showed that a two-pipe, direct-burial hot-water system using preinsulated pipe was the preferred approach. Inspection of the existing tunnels showed that there is not sufficient space in the majority of the tunnels to allow installation of hot-water piping. It was suggested, however, that the existing steam line could be used as either a hot-water supply or a return line. Use of the existing line would require only the burial of a single underground line in some areas. The estimated cost increase to implement this option was approximately $3.5 million (1979), which included heat-recovery plant, piping distribution, and building modifications. The analysis and detail costing of the hot-water distribution system were performed.[3]

Finally, the basic total energy system concept could incorporate both options (solid-waste energy input and hot-water distribution) and thereby achieve the highest energy savings, but with attendant higher capital costs. In the following section, the energy flow and system performance of the basic and alternate configurations previously described are discussed.

Installation considerations.- A number of factors must be considered in the design and installation of onsite power systems. The primary considerations are noise, thermal control, vibration, maintenance, and engine exhaust. However, a detailed analysis of these considerations is beyond the scope of this report. Engine noise abatement and vibration attenuation are achieved through design and selection of appropriate foundations and engineroom layouts.

[3]Central Heating and Cooling Plant Concepts and Analysis. Prepared by Bernard Johnson, Inc., for the NASA Johnson Space Center (contract NAS 9-14864), January 30, 1976.

Design practice for sound isolation is to completely enclose the engineroom. Concrete blocks (20 or 25 centimeters (8 or 10 inches) thick) filled with sand or poured concrete walls and concrete ceiling will reduce the sound pressure level outside the engineroom to acceptable levels for most facilities. Further reduction in sound pressure level within the room may be obtained by insulating the engineroom walls and ceiling with a layer of fiberglass covered with perforated wallboard.

The engine exhaust system should reduce the noise to an acceptable level; should not impose an excessive back pressure upon the engine (usually less than 3.7 to 6.2 kN/m^2 (15 to 25 inches water); should discharge the exhaust at a point not harmful or annoying to people or other facilities; and should dissipate a minimum amount of heat to the engineroom. Commercial heat-recovery units are available or may be fabricated to accommodate the preceding requirements for most diesel engines manufactured in the United States.

The engineroom must also be designed with sufficient ventilation to distribute and remove thermal energy radiated from the engine and its ancillary equipment. Typically, 6 to 8 percent of the fuel input energy to the engine is dissipated in the surrounding air. This ventilation is accomplished through use of induced draft or ventilating fans. It is also important to provide for an ample supply of cool, clean combustion air to obtain good engine performance. In large engine installations, the air intake is located at a point outside the engineroom but remote from sources of contamination such as engine exhaust, process fumes, etc.

The engineroom should be designed to facilitate engine maintenance and service. Individual engine manufacturers have specific requirements for floorspace between engines or between an engine and a parallel wall, overhead space, and space for auxiliary equipment. The same is true for exhaust-heat-recovery systems.

Figures 8 and 9 consist of a conceptual layout for a multiengine installation at the building 24 site. This layout (floorspace and ceiling height) was used for cost-estimating purposes.

<u>Energy analysis</u>.- The analyses of energy consumption for the alternate JSC utility systems were conducted primarily using the ESOP (Energy Systems Optimization Program) computer program.[4] This section includes a brief discussion of the portions of the ESOP used for this study, techniques used for energy analyses and comparisons, and the results of the energy analyses relating to the total energy option for JSC. The techniques described in this section are applicable to the total energy options and the engine-generator/electric-motor-driven chiller options considered in this study.

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[4] R. D. Stallings; S. L. Ferden; and E. S. Riley: Energy Systems Optimization (ESOP) User's Guide - Update IV. Lockheed Electronics Company, Inc., TM-4084, November 1974.

The ESOP consists primarily of a series of computer subroutines which predict heating, cooling, and water loads for a complex of buildings, and a series of subroutines which model equipment options that can be used to satisfy these loads. Also included in the program is the logic necessary to simulate the operation of equipment integrated into a variety of configurations to satisfy the predicted loads. The program includes a number of different equipment options for each major subsystem (i.e.; HVAC, power generation, solid waste, and wastewater) and simulates their operation when integrated into 25 different combinations. A conventional utility system is also simulated, and these options are compared for minimum energy consumption while satisfying common loads.

For the JSC IUS study, only the combination of subsystem equipment appropriate for the option under consideration was used. Also, because actual total loads were available, the entire building loads prediction section of the program was not required. (See section entitled "Utility Loads and Energy Consumption.")

The loads used for the evaluation of integrated utility system options were the same as those described earlier in the section entitled "Baseline JSC Utility System." Annual totals for the JSC loads and existing energy consumption are presented in figure 2. The loads were used in the ESOP as 24-hour profiles representing averages for summer workdays and holidays, winter workdays and holidays, and spring/fall workdays and holidays. Figure 3 consists of the six profiles used for cooling loads as well as minimum and maximum cooling-load profiles.

The profiles used for space-heating and domestic-hot-water loads are shown in figure 4. These two load components are shown together since they represent the total steam delivered to the distribution loop for these purposes, and they are indistinguishable after the steam leaves the central plant. The profile shown for maximum heating was used for the winter months, and the minimum heating profile was used for the remaining three seasons. It should be noted that these loads are not serviced by the total energy system because the 103.4-kN/m^2 (15 psig) steam produced by the power generating equipment was incompatible with the steam distribution loop, which operates at a pressure of 861.8 kN/m^2 (125 psig). Therefore, all recovered heat was used by the absorption chillers.

The electrical profiles shown in table II as the average for each season were used for the respective workdays. The profile shown for weekends and holidays was used for all seasons.

The ESOP requires inert performance data for the specific compression and absorption chillers and for the specific power generation equipment to be used. Single values for coefficient of performance (COP) of 4.0 for the electrically driven compression chillers and 0.6 for the absorption chillers were used as representative of good, commercially available equipment. Performance data for Delaval (Enterprise) engines were assumed to be representative of large, low-speed, IC engines. The specific fuel consumption (sfc) and heat-recovery data for the engine used are shown in table VI as a function of engine-generator rated load.

The results of the energy analyses are shown on energy flow diagrams and represent annual totals. The energy flow diagram for the total energy system configuration is shown in figure 10. Configuration 1 assumes that the existing 861.8-kN/m^2 (125 psig) steam distribution loop is used and that no change is made to the existing solid-waste disposal by landfill. The only available recovered heat is from the engine exhaust. However, utilization of this energy in absorption chillers provides approximately one-third of the JSC cooling load. The heat shown as exhaust on the flow diagram represents the heat that is lost from the engine by radiation, concentration, etc., and heat that is not recovered from the exhaust gases. The temperature of the oil-cooler and jacket-water heat is 352.59 K (175° F), and the heat is rejected in a dry-airblast heat exchanger. The boiler provides all of the site space heating and hot-water heating. It should be noted that all of the fuel heating values shown represent the higher heating value of the fuel. The prime-mover fuel is composed of approximately 10 percent fuel oil and 90 percent natural gas for the dual-fuel engines. This configuration results in an overall savings equal to approximately 38 percent of the site baseline energy consumption.

Figure 11 contains the results of the energy analysis for configurations 2 and 3. In these configurations, the existing 861.8-kN/m^2 (125 psig) steam distribution was used, but the heat content of the solid waste was used. Configuration 2 assumes incineration of the solid waste and use of the recovered heat to reduce the boiler load. Configuration 3 assumes pyrolysis of the solid waste and use of all the resulting gas by either the engine or the boiler. The two configurations are presented on a common flow diagram since the efficiency of both processes is 60 percent and use of the recovered energy is 100 percent in both configurations. The resulting energy savings from either of the two configurations is equal to 40 percent of the 1974 baseline consumption.

Figure 12 contains the results of the energy analysis for configuration 4. This configuration assumes that the existing landfill of solid waste is used, but that the steam distribution loop is replaced by a hot-water distribution loop at 352.59 K (175° F). The available heat from the engine water jacket and oil cooler will now replace the entire boiler load for average heat loads. It should be noted that a small boiler load will still exist on peak heating days, but the effect on annual energy demand will be minimal. This configuration results in a consumption savings equal to approximately 44 percent of the 1974 baseline.

Figure 13 is the energy flow diagram for configuration 5. This configuration assumes that the 352.59-K (175° F) hot-water distribution loop is used for space heating and hot water and that the solid waste is pyrolyzed to provide gas for the prime movers. The pyrolysis process is assumed to be 60 percent efficient, and the gas is assumed to be used at an efficiency of 100 percent in the prime mover. This configuration results in a savings of approximately 46 percent of the 1974 baseline energy consumption.

Cost analysis.- Estimates have been made for the JSC total energy concept for initial cost and annual operating and maintenance costs. These cost estimates were made in January 1976 dollars, and cost projections have been made to the year 2000. Emphasis has been placed on the estimates for years 1979 and later because such a system could probably not be operational before that time. Estimates of variations in fuel costs for different methods of handling the solid wate have been made without full consideration of variations in equipment and operation and maintenance (O&M) costs. Comparisons have been made to the existing electricity and gas/oil fuel costs. The cost of each option is summarized in table I.

The major cost comparison data for a total energy installation are the cost of fuel and purchased electricity. The projections made in a contractor study[2] have been used for these data. Initial equipment costs and annual O&M costs, excluding fuel, have been estimated in January 1976 dollars.

Initial costs for equipment and installation were based on trend data developed during the MIUS Program and on data from 1975 equipment and construction cost-estimating guides. Estimates have been included for design costs and general contractor profit and overhead; however, nothing has been included for contingencies and construction loan costs.

Operating and maintenance costs, excluding fuel, were based on data from the "1972 Report of Diesel and Gas Engine Power Costs," published by the American Society of Mechanical Engineers (ASME). Adjustments were made to these 1970 data to yield 1975 estimates.

Table VII contains estimates for the initial equipment outlay in January 1976 and in mid-1979 dollars. The estimate has been based on the component parts listed with the assumption that existing cooling towers, fuel storage equipment, and distribution systems will be used.

The variation in operating and maintenance costs is illustrated in figure 14. It should be possible to operate a new plant of the latest design and with the latest equipment at near the minimum costs of operating existing plants. Because year-by-year operating cost records were not available for several specific plants, it was judged reasonable and conservative to use average costs for this level of estimate. The ASME data are generally for powerplants without heat-recovery and air-conditioning equipment. No consideration has been given to the overall reduction of manpower that might result from combining the heating and cooling plant with the powerplant. There are possibilities for cost reduction in manpower, but a comprehensive estimate would require more extensive system definition and cost analyses than have been conducted.

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[2]Energy Outlook and Alternate Fuels Study - 1985-2000. Prepared by the Ralph M. Parsons Company for the NASA Johnson Space Center (job. no. 5494-1), November 1975.

In figure 15, the projected baseline-case cost of fuel and electrical power used in the analyses is shown. The cost of natural gas is predicted to exceed that of fuel oil before 1980; however, it has been assumed for the cost analyses and comparisons that JSC would continue to use natural gas.

Figure 16 contains a comparison of the cumulative dollar outlay for total energy system confirmation expenditures and expenditures for electrical power and natural gas for the existing boilers. The total outlay for the two cases is equal in approximately 13 years.

Engine-Generator/Electric-Motor-Driven Chiller

The provision for satisfying chilled-water requirements using reciprocating-engine-driven or diesel-electric-motor-driven compression chillers is shown in figure 17. The recovery of the high-grade water-jacket and exhaust-stack waste heat from the engines for use in absorption chillers is also implemented in this option. The prime movers would be located in a separate building near the existing faciltiy because of current plant size and structural limitations and excessive engine noise. This arrangement requires that the prime movers drive generators and that the resulting power be delivered to electric-motor-driven compressors in the main building. Possibly, a more efficient method would be to drive the compressors directly; however, a completely new facility would be required, and vibration problems inherent in direct-drive chillers would have to be solved.

Because seven 7034-kilowatt (2000 ton) compression chillers are presently installed, baseload requirements for the diesel/generator option were investigated for using this equipment size. Also, examination of the amount of recoverable engine exhaust-stack and water-jacket heat available for absorption cooling with this arrangement indicates a ratio of approximately 3.5 kilowatts (1 ton) of absorption for every 31.7 kilowatts (9 tons) of compression cooling that is generated.

A configuration using three 7034-kilowatt (2000 ton) engine-generator/electric-motor-driven chillers and one 2342-kilowatt (666 ton) absorption chiller could provide 23 443 kilowatts (6666 tons) of cooling. This configuration is shown in figure 17. One of the four remaining steam-turbine-driven chillers could be available for peak design-cooling days (27 174 kilowatts (7727 tons)) and another for backup and periodic maintenance flexibility. Cooling-tower requirements for the engine-driven chiller options are approximately 1.58 m^3/sec (25 000 gal/min), or only 60 percent of the present capacity. The following is a list of additional major equipment required for this option.

1. Three dual-fuel prime-mover/generator sets: 1611 kilowatts (1780 kilowatts peak)

2. Three gearbox assemblies: ratio of 4.8:1; rated at 1641 kilowatts (2200 horsepower)

3. Three exaust-heat-recovery boilers: temperature of exhaust entering boiler, 644 K (700° F); temperature leaving, 422 K (300° F); flow rate, 13 600 kg/hr (30 000 lb/hr); recovering approximately 791 kilowatts, $(2.7 \times 10^{6}$ Btu/hr) in the form of 394-K (250° F), 103-kN/m^2 (15 psig) saturated steam at rated load

4. One fuel oil storage tank: 75 700 liters (20 000 gallons)

5. One absorption chiller: 2342 kilowatts (666 tons)

6. Three 1678-kilowatt (2250 horsepower) synchronous electric motors

Optional configurations.- The only optional configurations considered in this case are energy recovery from JSC solid waste by either incineration or pyrolysis. As in the total energy case, heat recovered from solid-waste incineration was assumed to be used in the steam distribution system (space heating and hot water). For pyrolysis, the gaseous fuel produced may be injected into the air inlet of the dual-fuel engines (fumigation) as discussed in the section entitled "Total Energy Systems."

The energy analysis for the engine-generator/electric-motor-driven chiller was conducted in the same manner as that for the total energy options. The major changes that were made to the ESOP input to represent the engine-driven chiller option were the elimination of the site electrical load and changes in the engine performance data. The electrical load used consisted only of the compression chiller demand derived from the cooling load profiles shown in figure 17 assuming an overall COP of 4.0. The manufacturer's performance data for the engines used are shown in table VIII. It should be noted that the specific fuel consumption data have been adjusted to account for losses in the gearboxes. Because the engines can be ebulliently cooled, the water-jacket heat is recovered as steam at 103.4 kN/m^2 (15 psig), combined with the exhaust heat, and used by the absorption chiller.

The results of the energy analysis for configuration 1 are shown in figure 18. This configuration assumes that three electric-driven chillers and one absorption chiller are used for the base cooling load and that one existing steam turbine chiller is available as required for occasional peaking. Configuration 1 also includes solid-waste landfill. This configuration results in a consumption savings equal to approximately 24 percent of the total 1974 baseline. The fuel required by the central powerplant is based on a heat rate of 3.396 J/J (11 600 Btu/kWh).

Figure 19 is an energy flow diagram for the engine-driven chillers, configurations 2 and 3. These configurations assume pyrolysis or incineration of the solid waste. The two configurations are represented on a common energy flow diagram because the overall efficiency of either solid-waste system is 60 percent. In the pyrolysis system (configuration 3), the gas recovered is useful for either prime-mover or boiler fuel. In the incineration system (configuration 2), the recovered heat is used to reduce the boiler load. These configurations result in a savings equal to approximately 26 percent of the total baseline consumption.

Cost analysis.- The engine-generator/electric-motor-driven chiller drive concept provides for replacing the steam turbine drives on three chillers with electric motors and providing power for the electric motors with three 1611-kW engine-generator sets. Heat recovery from the water jackets and exhaust stacks provides 103.4-kN/m^2 (15 psig) steam for operation of a new 2342-kilowatt (666 ton) absorption chiller.

Estimates of the initial and annual O&M costs for this concept have been made. These cost estimates were made in January 1976 dollars, and projections have been made to year 1985. Comparisons have been made to the existing electricity and gas fuel costs. The cumulative costs of the existing utilities exceed those of this concept approximately 2 years after a 1979 installation; therefore, the concept appears economically attractive.

As in the total energy system cases, major cost trade parameters for installation include the cost of fuel and purchased electricity. Projections made by a contractor study[2] have been used for these parameters.

Initial costs for equipment and installation were based on trend data developed during the MIUS Program and on data from 1975 equipment and construction cost-estimating guides. Figure 20 is a diagram of capital costs for engine-generator sets and contains information on the data sources used.

Operating and maintenance costs, excluding fuel, were based on data from the 1972 Report on Diesel and Gas Engine Power Costs published by the ASME. Adjustments were made to these 1970 data to yield 1975 estimates.

Table IX contains estimates for the initial equipment outlay in January 1976 and in mid-1979 dollars. The estimate has been based on the component parts listed and the assumption that existing cooling towers and distribution systems will be used.

Figure 21 is a comparison of the cumulative outlay for the engine-generator concept expenditures and expenditures for existing electrical power and natural gas for the boilers. The total outlay for the two conditions is equal in approximately 2 years.

Figures 22 and 23 are site and floor plans for the installation of major additional equipment for the engine-driven chiller option. Engine maintenance considerations would be essentially the same as discussed in the section entitled "Total Energy Systems."

[2] Energy Outlook and Alternate Fuels Study - 1985-2000. Prepared by the Ralph M. Parsons Company for the NASA Johnson Space Center (job. no. 5494-1), November 1975.

Electric-Drive Chiller Option

In the electric-drive chiller option, steam turbines would be replaced with electric motors for driving chillers and evaporators would be modified to provide for maximum chiller output. Many variations are possible in the modification of the system to electric-motor-driven chillers; examples are the number and type of chillers to be converted, ancillary equipment, and modifications to the evaporators.

The system modifications selected for this option for this study consist of converting the three turbine-driven York centrifugal-chiller units to electric drive and converting the three-pass evaporators on each of the seven chillers to two-pass evaporators. The modifications for the electric drive to the York chillers include the installation of speed-increased gearboxes, changes to the York chiller unit foundations, and piping changes to the York condensers.

Also included is the upgrading of the seven chilled-water pumps required to provide the increased chilled-water flow for the two-pass evaporators. This configuration also includes changes to the cooling-tower fans necessitated by the lower heat-rejection requirements resulting from the deletion of the steam turbine drives. The selection of the three York chillers for electric drive and the recommendation to use only three chillers are based on the following factors.

1. The York compressors are provided with prerotation inlet vanes that allow economical load modulation.

2. The brake horsepower per ton-hour for the York compressors is lower than that for the Carrier compressors.

3. Based on the actual energy production load requirements for 1974, chilled-water generation was 26.8 percent less than that for the same period of 1972.

4. The operation of building 48 chillers for the building 30 mission operations wing and administrative wing at high load demands would significantly reduce the chilled-water demand on the plant and result in the need for only three units in operation.

5. The recirculting of the chiller evaporators to allow for maximum loading of all plant chillers was considered in determining that the initial phase should be included on the three York units. Flow modulation at all load conditions would remain a mandatory operating procedure and is also an important factor in determining the number of units to select for this modification.

Figure 24 is a simplified schematic of the electric-drive chiller option.

Energy analysis.- As a baseline for comparing energy consumption, actual CHCP production data for calender year 1974 were used. (See fig. 2.) Because adequate natural gas flow measurement devices were not available during this time period, the following rationale was used in calculating natural gas consumption from steam and chilled-water production logs.

Average steam consumption per unit of refrigeration (derived from log sheets and test runs) was 0.559 kg/MJ (15.6 lb/ton-h). Steam supplied to the turbines at 2758 kN/m^2 (400 psig) and 589 K (600° F) leaves the turbines at 13.5 kN/m^2 (4 inches mercury) and 325 K (125° F), resulting in a specific enthalpy change Δh of 2.8 MJ/kg (1213 Btu/lb). Assuming a boiler efficiency of 75 percent, energy to the boilers is

$$\frac{0.559\ \text{kg/MJ} \times 2.8\ \text{MJ/kg}}{0.75} = 2.0869\ \text{J/J} \tag{1a}$$

or

$$\frac{15.6\ \text{lb/ton-h} \times 1213\ \text{Btu/lb}}{0.75} = 25\ 230\ \text{Btu/ton-h} \tag{1b}$$

Using a conversion factor of 38.39 MJ/m^3 (1031 Btu/ft^3), natural gas comsumption becomes

$$\frac{2.102\ 499\ 9\ \text{MJ/MJ}}{38.388\ 334\ \text{MJ/m}^3} = 0.054\ 769\ 2\ \text{m}^3\text{/MJ} \tag{2a}$$

or

$$\frac{25\ 230\ \text{Btu/ton-h}}{1\ 031\ 000\ \text{Btu/10}^3\ \text{ft}^3} = 0.024\ 47 \times 10^3\ \text{ft}^3\text{/ton-h} \tag{2b}$$

To supply 25 304.5 megajoules (2000 ton-hours) of refrigeration using a 5626.8-megajoule (2095 horsepower-hour) electric motor which drives a chiller and assuming a 96-percent motor efficiency and a 98-percent gearbox efficiency, the energy ratios are

$$\frac{\dfrac{5626.8\ \text{MJ}}{25\ 304.5\ \text{MJ}} \times \dfrac{26\ 856\ \text{J}}{26\ 856\ \text{J}}}{0.96 \times 0.98} = 0.2363\ \text{J/J} \tag{3a}$$

or

$$\frac{\frac{2095 \text{ hp-h}}{2000 \text{ ton-h}} \times \frac{0.746 \text{ kWh}}{\text{hp-h}}}{0.96 \times 0.98} = \frac{0.8306 \text{ kWh}}{\text{ton-h}} \tag{3b}$$

Using an average powerplant heat rate of 3.395 J/J (11 Btu/kWh) and 38.39 MJ/m^3 (1031 Btu/ft^3), natural gas consumption by the powerplant is

$$(0.236\ 34 \text{ MJ}_e/\text{MJ}_t)\ (3.395 \text{ MJ}_t/\text{MJ}_e)\ (0.026\ 049\ 6 \text{ m}^3/\text{MJ}_t) = 0.020\ 901\ 5 \text{ m}^3/\text{MJ}_t \tag{4a}$$

or

$$(0.8306 \text{ kWh/ton-h})(11\ 600 \text{ Btu/kWh}) \times \frac{10^3 \text{ ft}^3}{1.031 \times 10^6 \text{ Btu}} = 0.009\ 345 \times 10^3 \text{ ft}^3/\text{ton-h} \tag{4b}$$

where the subscripts e and t denote electrical and thermal energy, respectively. Therefore, to produce 137.8 GWh (39.2 × 10^6 ton-h) of cooling by electric-drive chillers, 117.4 terajoules (32.6 × 10^6 kilowatt-hours) of additional electrical energy is required. The electric utility company would then supply a total of 567.0 terajoules (157.5 × 10^6 kilowatt-hours) and would use 1925 terajoules (1826 × 10^9 British thermal units) annually. The annual JSC natural gas consumption would be only 283 terajoules (268 × 10^9 British thermal units) used for space heating and hot-water heating. The total annual energy consumption would be 2208 terajoules (2094 × 10^9 British thermal units), or 22.6 percent savings over the baseline year. Figure 25 is the energy flow diagram for this case. The actual savings with only three electric-drive chillers would be approximately 21.6 percent because peak loads (approximately 21 000 kilowatts (6000 tons)) would be met by steam-turbine-driven chillers.

Cost analysis.- Capital cost estimates (table X) were made in applicable end-of-year dollars. An escalation rate of 10 percent per year compounded to mid-1978 was used to determine capital cost invested when equipment becomes operational. Capital cost estimates assumed that additional power cables and a fourth complete chiller/motor package would be required. Table XI contains a comparison of energy costs incurred using electric-motor-driven chillers and using the existing gas-fired steam-turbine-driven chiller system.

Figure 26 shows the cumulative total site fuel costs from 1979 to 2000 for the existing utilities system and for the system using electric-motor-driven chillers. The total capital cost of the electric-motor-driven chillers ($2.5 million) is added to the cumulative fuel costs for that option. Crossing of the two curves at mid-1980 indicates a 1.4-year payout period. By the year 2000, the electric-motor-driven chillers will have saved approximately $63 million.

Combination Steam-Turbine-Driven/Compression Chillers and Absorption Chillers

In the existing JSC heating and cooling plant, steam turbines are the prime movers for the centrifugal chillers. The turbines convert only a fraction of the thermal energy present in the inlet steam to mechanical power. The steam leaving the turbine contains a large amount of energy, which is removed by the steam condenser and rejected to the atmosphere by the cooling tower. Combination centrifugal/absorption chiller systems can use this energy by passing the turbine exhaust steam to absorption machines. The absorption machines serve as condensers for the turbines and can use almost 100 percent of the exhaust steam latent energy.

Steam expands from an initial pressure to a design back pressure in a turbine, with attendant loss in temperature. Thus, the amount of energy that can be converted tc mechanical power depends on the difference of the heat content, or enthalpy, of the steam between the two conditions. When a turbine exhausts steam to an absorption unit, the pressure of the exhaust should be at least 83 kN/m^2 (12 psig), the inlet design pressure of the chiller. The current theoretical performance of the existing system, expressed as steam consumption per unit of cooling capacity, is 0.337 kg/MJ (9.4 lb/ton-h) based on the following parameters: 2758-kN/m^2 (400 psig), 533.2-K (500° F) steam exhausting to 13.2-kN/m^2 (1.92 psia), and 70-percent turbine adiabatic efficiency. The theoretical performance of the combination, using turbine exhaust to drive the 83-kN/m^2 (12 psig) absorption units, results in an adiabatic efficiency of approximately 60 percent and an overall performance of 0.329 kg/MJ (9.17 lb/ton-h). This very small advantage (0.008 kg/MJ (0.23 lb/ton-h)) is almost inconsequential (i.e., total energy savings would be less than 1 percent) and does not justify the significant expenditure for altering the steam turbines and for addition of absorption chillers and support equipment.

Solid-Waste Management/Energy Recovery

Two systems for conversion of JSC solid waste into energy were studied: an incineration system with heat recovery in the form of steam and a pyrolysis system that produces a fuel gas to be used as boiler or engine fuel.

<u>Incineration systems</u>.- The proposed incineration system consists of the following major equipment items.

1. C-760 Consumat incinerator - 70-kW/day (20 ton/day) capacity

2. Heat-recovery unit

3. Automatic loader

4. Automatic ash-removal system

The cost of the equipment is estimated at $250 000. A building with approximately 185.8 square meters (2000 square feet) of floor space is required to enclose the waste dumping area. At $323/m^2$ ($30/ft^2$), the cost would be $60 000; the total capital cost would be $310 000. The incinerator and heat-recovery equipment would be located outside. The steam produced would be supplied to the existing steam system for the Center. Energy would be recovered at 60 percent efficiency and would amount to approximately 64.3 terajoules (61×10^9 British thermal units) annually including steam produced by the supplemental fuel required by the incinerator. Supplemental fuel would be required approximately 12 hr/day, 6 days/week, which would necessitate two shifts per day. It is estimated that 1.5 men/shift could successfully handle the operation.

The Consumat incinerator is preferred because it is an off-the-shelf item in the capacity range required. Other incinerators available are as follows.

1. Kelly-Hoskinson incinerators have capacities as great as approximately 635 kg/hr (1400 lb/hr). Two such incinerators would be required to handle the JSC solid waste. The energy recovery efficiency is approximately the same as for the Consumat. The Kelley-Hoskinson incinerator is a starved-air device similar to Consumat and could satisfactorily be used as an alternate.

2. A Clean-Air-Ator excess-air-type incinerator was tested in the MIST. Problems with temperature control caused slagging in the primary chamber. Loading problems were also encountered but could probably be solved with a different loader.

3. The minimum economic size of the water-wall-type incinerator is near 108.9 Mg/day (120 tons/day); therefore, it does not apply to JSC.

Pyrolysis systems.- The pyrolysis system proposed for use in the JSC IUS consists of the following equipment.

1. Urban Research and Development Corporation (URDC) pyrolysis reactor - 18-Mg/day (20 ton/day) capacity

2. Loading system

3. Residue removal system

The capital cost for the preceding equipment is estimated at $340 000. As in the case of the incinerator, a building with 185.8 square meters (2000 square feet) floorspace would be required to house the dumping area. The estimated cost for the building is \$60 000. The total capital cost would be \$400 000. The fuel gas produced by the pyrolysis system would be either blended with natural gas or fired directly into an existing or new boiler to produce steam for the JSC system. The amount of energy in fuel gas produced annually is approximately 60 terajoules (57×10^9 British thermal units). No supplemental fuel is required. The pyrolysis unit will be operated 24 hr/day, 6 days/week. Three shifts with 1.5 men/shift should provide the needed manpower for operation of the system. The following available pyrolysis systems were evaluated.

1. URDC system - This system uses air for partial oxidation of solid waste. The reactor produces a low-energy gas (less than 5.58 MJ/m^3 (150 Btu/ft^3)) and a solid (slag) residue. The URDC system is simple and has potential for various size installations. Use of the product gas is limited; however, the gas should be suitable for boiler fuel.

2. Union Carbide system - This system uses pure oxygen for the partial oxidation process, which necessitates construction of an oxygen production plant or purchase of oxygen. The smallest economically feasible plant will handle approximately 181 Mg/day (200 tons/day) of solid waste; therefore, this system is not suitable for the JSC IUS.

3. Barber-Colman Company system - This system is based on lead-bath transport pyrolysis and has been demonstrated and tested on a small-scale version (31.8 kg/hr (70 lb/hr)) under NASA contract. The gas produced has a heating value near 18.6 MJ/m^3 (500 Btu/ft^3), and the system efficiency is approximately 60 percent. The process allows recovery of inorganic materials (glass and metals) if desired. This system requires further testing on a larger scale for determining feasibility.

4. Monsanto system - The capacity of this system must be greater than 181 Mg/day (200 tons/day) for economical operation. Fuel oil is required to sustain the process.

Tables XII and XIII contain costs for incineration and pyrolysis, respectively. Because JSC has a year-round demand for steam, incineration with heat recovery is the most cost-effective system. This conclusion is primarily based on two factors: (1) the cost of incineration with heat recovery is 30 percent less than the cost of pyrolysis and (2) two-shift operation with the incinerator will dispose of the waste, whereas three shifts are required for pyrolysis. At \$1.42/GJ (\$1.50/10^6 Btu), simple payout for incineration with heat recovery is less than 10 years. Pyrolysis is not economically feasible until the price of natural gas exceeds \$1.42/GJ (\$1.50/10^6 Btu). For this specific application, incineration with heat recovery provides the best approach.

Figure 27 is an incinerator system flow diagram. Figure 28 is a sketch of the Consumat incinerator. Figure 29 is a sketch of the URDC pyrolysis system sized for approximately 18.1 Mg/day (20 tons/day) capacity.

JSC Water Management Considerations

The wastewater from JSC is presently collected and conveyed by an underground piping system to the Clear Lake City Water Authority (CLCWA) wastewater treatment facilities, where it is treated to meet State quality standards and discharged to Horsepen Bayou and from there to Clear Lake. As part of the contract with CLCWA, JSC has the right to take the treated wastewater and return it to JSC for any desired reuse. Three areas of significant nonpotable water usage at JSC have been identified and analyzed as potential areas for wastewater reuse; however, no detailed cost analysis has been performed.

First, JSC used approximately 87 064 cubic meters (23×10^6 gallons) of potable water per year for irrigating the grass areas. The treated wastewater from CLCWA could be used for this purpose with proper planning and precautions to prevent the possible usage of the water for potable purposes. One disadvantage of this reuse application would be the conversion of the existing extensive sprinkler system from potable water connections to treated wastewater connections or the installation of a totally new distribution system. The only additional treatment for this reuse would be disinfection.

Second, JSC uses approximately 28 390 cubic meters (7.5×10^6 gallons) of potable water per year for building 24 boiler makeup. The treated wastewater from CLCWA could be used for this purpose with additional treatment. The quality of water needed for the boilers would require that both suspended and dissolved solids be removed from the treated wastewater. Also, additional piping would be required to deliver the treated wastewater from CLCWA to JSC.

Third, the largest nonpotable water usage at JSC is the 632 164 cubic meters (167×10^6 gallons) of potable water per year used for building 24 cooling-tower makeup. The treated wastewater from CLCWA could be used for this purpose with less treatment than would be required for the boiler makeup. However, additional treatment to remove residual suspended solids and dissolved organics would be required. Solids would be removed through filtration. The quantity of blowdown water to be treated would be approximately 50 percent of the 632 164 cubic meters (167×10^6 gallons) per year of cooling-tower makeup water. The remaining 50 percent of makeup water is estimated to be lost by evaporation. Additional piping would also be required to deliver the treated wastewater from the CLCWA plant to JSC.

The JSC has been directed to stop using ground water and has contracted with CLCWA to use surface water supplied by the City of Houston. Use of surface water was begun by JSC on June 17, 1976. The previously used water wells are being maintained on standby for emergency use only. Surface water is priced sufficiently low that the aforementioned water reuse possibilities would not be economical. This same conclusion was reached by an engineering company in a study performed for JSC in 1971.[5]

[5]Study of MSC Wastewater Collection System for Discharge to Facility of Clear Lake Water Authority and Feasibility Study for Recycle of Treated Wastewater to MSC. Prepared by Bovay Engineers, Inc., 1971.

Energy Storage Concepts

A typical application of energy storage in JSC-type facilities is the storage of excess heat energy for use as supplemental heating during periods when extra heating is required. Investigation of the availability of excess heat in the options studied in this report showed that all recovered high-grade waste heat was used for absorption air-conditioning with no excess; therefore, heat storage cannot be a practical consideration here.

Chilled-water storage may be used to make better use of chiller capacity during offpeak periods and thereby to take advantage of better chiller performance and sometimes better electric rates while keeping installed capacity near minimum. This application was investigated for the electrically driven chiller option using the three 7034-kilowatt (2000 ton) centrifugal chillers.

Figure 3 shows that 13 hours of each average summer workday exceed the 21 100-kilowatt (6000 ton) electric chiller capacity. For each average summer workday, the 13 hours represent 117.4 gigajoules (9272 ton-hours) beyond the hourly capacity, whereas 114.4 gigajoules (9041 ton-hours) on the same day would be available for storage during the hours when chilled-water requirements are less than 21 100 kilowatts (6000 tons). For storage sizing, the workday deficiency for the 5-day workweek should be totaled with the 117.4 gigajoules (9272 ton-h) and approximate storage losses. This would allow sufficient capacity to begin the 13-hour period at the beginning of the workweek and never deplete the storage capacity on the average summer days until the weekend arrived and replenishment began. Based on this procedure, the storage capacity should be approximately 139 gigajoules (11 000 ton-h) including approximate losses. The existing steam-turbine-driven chillers would be used to supply requirements beyond the 1323 gigajoules (144 000 ton-h) daily requirement which is derived from 24 hours operation at 21 100-kilowatt (6000 ton) capacity. Considering the size of a 139.3-gigajoule (11 000 ton-h) storage tank using the sensible heat of water and a temperature change ΔT of 8.9 K (16° F) (277.6 to 286.5 K (40° to 56° F)), a volume of 3736.4 cubic meters (131 948 cubic feet or 986 971 gallons) would be required.

Based on 1980 energy costs, the baseline JSC consumption rates indicate charges of approximately $\$0.104/m^3$ ($\$2.95/10^3\ ft^3$) of natural gas and \$0.006/MJ (\$0.023/kWh) electricity. Although charges are not linear for the ranges of this study, a linear assumption will allow legitimate comparative values.

Plant operation logs for the baseline year show 26 920 cubic meters ($950\ 655 \times 10^3$ cubic feet) of natural gas used for the chiller steam turbines and 137.75 gigawatts (39.17×10^6 tons) of cooling generated. Based on $\$0.104/m^3$ ($\$2.95/10^3\ ft^3$), the cost was approximately \$0.02/kW (\$0.07/ton) for the turbine-driven chillers. The electrically driven chillers produce approximately 4.22 kilowatts (1.2 tons) cooling per kilowatt of electricity. (See the section entitled "Electric-Drive Chiller Option.") At a rate of \$0.006/MJ (\$0.023/kWh), an assessment of \$0.0055/kW (\$0.0192/ton) for electrically driven chillers is applicable.

Based on the annual JSC cooling load and 100-percent steam turbine chillers, the annual cost of cooling would be approximately \$2 800 000. The 21 100-kilowatt (6000 ton) electric-drive chiller option, which uses steam-turbine-driven chillers for the hours that exceed a 21 000-kilowatt (6000 ton) load, would reduce the annual cost to \$776 000. Introducing thermal storage as discussed would allow total operation with electrically driven chillers and the annual costs would be further lowered to \$755 000. These calculations do not indicate the added advantages of increased chiller performance gained by shifting a portion of the cooling to nighttime operations nor do they reflect probable bulk electrical rate advantages gained from peak shaving.

Although a storage tank capable of containing almost 3785 kiloliters (1 000 000 gallons) of water can provide a variety of options, the cost of a cylindrical, steel-reinforced, concrete tank of this size buried below the ground represents an additional investment of approximately \$80 000 in 1975 dollars or \$118 000 in 1980 dollars (10-percent escalation rate). Allowing another \$6000 (1975 dollars) for extra plumbing, pumps, and controls, the total cost of \$126 000 represents an annual savings of \$21 000 and an investment with a payback period of 6.0 years over the no-storage, electric-drive option.

Solar Energy Applications

The recent national emphasis on the use of solar energy to aid in lowering fossil fuel requirements indicates possible application in JSC-type facilities. The major utility options for use of current solar energy technology in JSC-type facilities provide energy assistance for the supply of domestic hot water (DHW), space heat, and absorption cooling. Solar dehumidification using the desiccant process is also considered in the following discussions. After this study, JSC personnel initiated a solar energy project wherein solar heat will be used for dehumidification control (conditioned-air reheat) in the building 30 computer facilities. This application was not considered in this study.

<u>Domestic-hot-water option</u>.- Currently, the most practical application of solar energy in the utility system is by integration into the DHW system, which is illustrated in figure 30. Normally, this system requires temperatures in the 333.2- to 344.3-K (140° to 160° F) range, and standard flat-plate collectors can operate effectively supplying this level for a reasonable portion of a clear day. Figure 31 shows the general construction of a flat-plate collector. Figure 32 illustrates the relative efficiencies of a representative flat-plate collector on a clear day at several water outlet temperatures. Another major advantage in using solar collectors to supply the energy for DHW is that, generally, the daily demand total is nearly constant throughout the year for most applications and results in better utilization of the collector system investment.

The JSC has a very small DHW energy requirement. With the exception of very few buildings, the only requirement for DHW is for restrooms and janitorial services. Therefore, supplying these buildings with individual collector systems, however small, would not justify the hardware expense. Solar assistance from a central collector field supplying these buildings through the utility tunnel system is a further consideration. Solar collectors might then reasonably supply a portion of the baseload.

A few JSC buildings have relatively high DHW demands; examples are the building 3 and 11 cafeterias. Currently, the cafeterias use steam generators from the distribution loop to supply hot water at a temperature of approximately 344 K (160° F). Most of this water is then heated for dishwashing. Measurements performed by the Center Operations Directorate, Engineering Division, indicate that the building 3 cafeteria uses approximately 37.85 cubic meters (10 000 gallons) of DHW each workday, and building 11, approximately 30.28 cubic meters (8000 gallons) of DHW each workday. These values equate to daily energy consumption of 8.65 gigajoules (8.2×10^6 British thermal units) and 6.96 gigajoules (6.6×10^6 British thermal units), respectively, using a ΔT of 55.6 K (100° F).

Figure 33 illustrates the range of insolation on a horizontal surface on clear and on average winter and summer days in the Houston area. The substantial decrease in insolation on the average days is primarily due to the high humidity and cloud cover and to other particulates. Figure 34 shows the effect of tilting the collector surface to 29.5° to the horizontal, the latitude of Houston. This effect tends to converge the daily insolation profiles of each month to similar levels for year-round optimization.

Figure 35 represents solar collector performance data for a standard flat-plate unit as supplied by a major manufacturer. Several options are available in the advertised line, and, following the manufacturer's recommendations for the Houston area, a collector with collection tubes on 14-centimeter (5.5 inch) centers and single glass glazing was selected. More efficient collectors having selective surfaces, extra glazings, and more closely spaced collection tubes could be purchased, but the costs are significantly higher.

Using a ΔT of 55.6 K (100° F) in figure 35, it can be seen by extrapolation that an input of more than 441 W/m^2 (140 $Btu/hr \cdot ft^2$) is necessary to achieve measurable output. The families of curves actually extend down to only 10 percent efficiency because line losses and pumping power make output at lesser efficiencies impractical. At 10 percent efficiency, it is necessary to receive at least 504 W/m^2 (160 $Btu/hr \cdot ft^2$) input to realize a dividend.

From figure 34, it can be seen that the hours above 504 W/m^2 (160 $Btu/hr \cdot ft^2$) from 9:30 a.m. to 2:30 p.m., (a total of 5 hours) obviously encompass a very small portion of the available energy. From hourly totals above 504 W/m^2 (160 $Btu/hr \cdot ft^2$), calculated for average insolation days each month, the annual daily mean is only 40 W/m^2 (305 $Btu/day \cdot ft^2$). To supply the total DHW requirements of building 3, a collector area of 2508.4 square meters (27 000 square feet) would be required. This area is more than 28 percent greater than the cafeteria roof area. Under the specified conditions, building 11 would require 2006.7 square meters (21 600 square feet) of collection on its 1521.5-square-meter (16 377 square foot) roof. Based on a collector cost of \$86.11/m^2 (\$8.00/ft^2) (currently a relatively low price) plus an approximate 25 percent for plumbing hardware and installation, the payback period is 43 years.

The most logical improvement to the DHW system design is shown in figure 30. If the solar collectors were used only to preheat the DHW, a much greater portion of the daylight period could be utilized at significantly higher collector efficiencies. Assuming a ΔT of only 27.8 K (50° F), figure 35 shows the usable solar input energy starting at approximately 189 W/m^2 (60 $Btu/hr \cdot ft^2$). Figure 34 indicates almost twice as many collection hours and several times more collectable energy. The daily average for the year is increased to 106.8 W/m^2 (813 $Btu/day \cdot ft^2$), and the collector fields required are 947.6 square meters (10 200 square feet) for building 3 and 752.5 square meters (8100 square feet) for building 11. Because only half the energy is saved for each system, the payback period is calculated to be approximately 32 years.

The payback periods in both considerations are not unusual in the relatively new solar energy field. As a result, only a proportionally few applications are proving cost-effective. However, assuming that fuel costs will continue to increase and that the cost of solar equipment will decrease because of mass production, design improvements, and competition, solar energy utilization will become fairly commonplace, as it is already in several other countries.

Space heating.- Because of the very seasonal demand, space heating using current solar collection methods has even fewer applications than supplying DHW in the retrofit market. This drawback is even more apparent in southerly locations such as Houston. The lower temperatures associated with solar collector output also dictate either higher surface areas in the heat exchanger units in the spaces to be heated or higher flow rates. Either modification requires significant additional expense. Because of these factors, space-heating assistance by solar collectors on an individual building basis is not a realistic option at JSC.

Another design approach would be to locate the solar collectors in a large central solar collector field and supply heating to individual buildings by a pumped hot-water distribution system. Even this approach, however, is not cost-effective, as is indicated by the example in the following paragraphs.

The hot-water distribution system would cost approximately $1.75 million (1979 dollars).[3] To meet the annual space-heating and DHW load of 168.7 terajoules (160×10^9 British thermal units) (fig. 2), an hourly average of 57.99 gigajoules (0.055×10^9 British thermal units) for 8 hours is required each day of the year. According to heating-load data, this annual demand is relatively constant. Figure 34 indicates an average annual solar input of approximately 473 W/m^2 (150 $Btu/hr \cdot ft^2$) for 8 hours daily. With a 44.4-K

[3]Central Heating and Cooling Plant Concepts and Analysis. Prepared by Bernard Johnson, Inc., for the NASA Johnson Space Center (contract NAS 9-14864), January 30, 1976.

(80° F) ΔT, figure 35 indicates a collector efficiency of approximately 20 percent at this input rate and a collector output of only 78.8 W/m^2 (25 Btu/hr•ft^2). Therefore, the total area required is approximately 204 400 square meters (2.2 × 10^6 square feet), which is 16.11 megawatts (0.55 × 10^9 Btu/hr) divided by 78.8 W/m^2 (25 Btu/hr•ft^2).

At $86/m^2 ($8/ft^2), the solar collector cost would be $17.6 million, not including a load-leveling storage system and oversizing requirements to achieve input-output averaging. The total cost, including hot-water distribution, would be approximately $19.35 million. The payback time for this investment would be 32 years, based on 1980 natural gas prices and the baseline-year consumption rate.

Absorption cooling.- Central plant size absorption chillers nominally operate on 82.7 to 96.5-kN/m^2 (12 to 14 psig) steam or 388.7 to 394.3-K (240° to 250° F) hot water. Somewhat degraded performance is occasionally acceptable with slightly lower energy levels, but the component heat-transfer surfaces and concentrations of the refrigerant and absorbent are engineered for a limited range of inlet conditions.

Solar energy systems can provide sustained temperatures in the working level of the absorption units by use of solar concentrators. Numerous concentrator designs have been conceived, but few have been tested and proven desirable for long-term absorption air-conditioning. The following are major drawbacks to the use of concentrators.

1. Concentrators can collect only direct sunlight; thus, the diffused radiation that represents more than 30 percent of the insolation in most parts of the country is lost. Flat-plate collectors absorb both direct and diffused radiation.

2. Most concentrators require periodic adjusting or constant-tracking mechanisms to maintain optimum orientation.

3. Concentrators cost much more than flat-plate collectors.

The emphasis in adopting absorption cooling to solar energy has been on modifying the absorption chillers to operate at lower temperatures. Currently, only one manufacturer is known to offer a commercially available unit; these were expected to be available in late 1976 in the 17.6- to 87.9-kW (5 to 25 ton) range. Cost is projected to be many times higher than for the equivalent size compression unit.

Emphasis continues on improving the performance of the flat-plate collectors. Better, less expensive glazings and selective surfaces should lower costs and raise efficiencies; thus, absorption cooling is expected to become feasible in the near future. Once the seasonal demands of space cooling can be adequately assisted with solar energy, the equally seasonal space-heating loads furnished by solar energy become more attractive because the same solar system can then be utilized on a year-round basis.

Solar dehumidification using the desiccant process.- The JSC Urban Systems Project Office contracted a comprehensive study of solar energy applications in integrated utility systems.[6] As a part of that study, a detailed evaluation was made of solar dehumidification systems using desiccants. This evaluation is pertinent to JSC solar energy applications and is therefore presented in the following paragraphs.

The desiccant process: Desiccant dehumidification systems use materials (desiccants or sorbents) that are capable of attracting and removing water from an airstream. The desiccant may be either liquid, such as a glycol compound, or solid, such as lithium chloride or molecular sieve material. The water may be removed either by absorption, which involves a physical or chemical change in the desiccant (as in the case with materials such as glycol or lithium chloride), or by absorption, which involves no physical or chemical change in the desiccant but generally depends on surface effects (as is the case with molecular sieve material).

All of the previously mentioned materials remove water by reversible mechanisms and therefore can be regenerated. Although some nonregenerative systems, such as those using hygroscopic salts, have value for some applications (generally for small batch process operations), they are not suitable for space conditioning.

The basic regenerative dehumidification cycle includes two elements: sorption and regeneration. Sorption is removal of water vapor from the airstream into the desiccant. This process generates heat, the major portion of which is the heat of condensation of the water vapor. Means of heat removal must be provided either with an external coolant or by accepting the rejected heat as a temperature rise in the process air. The lower the temperature of the sorption process, the lower the attainable air dewpoint. Regeneration is the removal of water from the desiccant. Regeneration must be accomplished by heat addition in the form of heating coils and/or a warm regenerative gas stream. The degree of sorbent drying is proportional to the regeneration temperature.

The basic ways in which desiccant systems can be applied to air-conditioning include:

1. Improving comfort through lowering relative humidity alone.

2. Serving as a preconditioning step that lowers the latent heat load on the refrigeration-type air-conditioner - this preconditioning allows the refrigeration-type air-conditioner to operate at both a reduced load and at higher evaporator temperature.

3. Achieving cooling as well as dehumidification by overdrying the air and then rehumidifying it to achieve the desired temperature and relative humidity.

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[6]Feasibility Study of Solar Energy Utilization in Modular Integrated Systems. Prepared by Arthur D. Little, Inc. (NASA contract NAS 9-14524), June 30, 1975.

4. Using desiccant enthalpy exchangers to accomplish latent, as well as sensible, heat recovery between inlet ventilation air and discharge air.

A COP for regenerative desiccant systems can be defined as the ratio of latent heat removal to regenerative heat addition. As a theoretical limit, one might expect the COP to approach unity; i.e., regenerative heat addition just equal to heat of evaporation.

Solar desiccant dehumidification performance characteristics: Solar desiccant dehumidification systems have the following characteristics.

1. They are probably only practical for individual units located at each building, because their use in a central facility would necessitate the transfer of air or of concentrated liquid desiccant from the central facility.

2. They can provide summer use of solar collector design for winter heating.

3. If used only for latent load (or a portion thereof), the collector will generally cover a reasonably small portion of roof area - perhaps comparable to heating area requirements for commercial buildings.

4. Prior dehumidification allows the occurrence of sensible cooling at a higher chilled-water temperature (or evaporator temperature) and thereby improves the COP of the primary chiller (compression and/or absorption).

5. The COP will generally be comparable to that of absorption machines.

6. Some types of desiccant systems can be designed to operate at temperatures lower than those required for absorption machines.

7. The best applications appear to be those in which latent loads are a large fraction of the total air-conditioning load and in which cooling is required throughout the year.

8. Liquid systems require cooling towers and are probably best matched to larger buildings.

A summary of some characteristics of the various desiccant systems is presented in table XIV.

Evaluation/cost analysis of desiccant systems: For individual buildings, the solar desiccant system must compete on the basis of reducing the usage and installed capacity of centrally supplied chilled water. The installed capacity, including prime movers, electric chillers, and associated chillers, is assumed to be sized to match the peak summer cooling requirement.

The preceding rule implies that the solar dehumidification system would not heat domestic hot water (at any times other than peak cooling periods) or contribute to space heating. It must be a "cooling only," or "dehumidification only," system. Therefore, to receive reasonable use, the solar dehumidification system must be applied to a building that has a fairly uniform dehumidification load during the entire year. For cost analysis, the following incremental cost and cost saving associated with the solar desiccant equipment were neglected.

1. The incremental cost impact of replacing central electric or absorption chiller capacity with individual desiccant units is an unimportant element here. Considering a collector area of 3.96 m^2/kW (150 ft^2/ton) (a reasonable approximation for actual sizing), a differential cost of \$28.43/kW (\$100/ton) of cooling would only amount to a cost of \$7.21/m^2 (\$0.67/ft^2). This is only 6.7 percent of the total solar system cost of \$107.64/m^2 (\$10/ft^2).

2. The reduction in installed cost of the IUS prime mover and generator could be important and should be considered in further studies.

Numerical example: The cost of the solar heat collected can be calculated as

$$\mathrm{SHC} = \frac{\frac{C}{A}}{\bar{n}\left(\frac{Qi}{A}\right)}$$

where the annual collection efficiency $\bar{n}$ = 40 percent (from fig. 35), the system installed cost C/A = \$86/m^2 (\$8/ft^2) (optimistic), and annual insolation Qi/A = 6364.56×10^6 J/m^2·yr or 0.56×10^6 Btu/ft^2·yr (from fig. 34). Therefore,

$$\mathrm{SHC} = \frac{\$86}{(0.4)\ (6364.56 \times 10^6\ \mathrm{J/yr})}$$

$$= \$0.034/10^6\ \mathrm{J{\cdot}yr} \qquad (5a)$$

or

$$\mathrm{SHC} = \frac{\$8}{(0.4)\ (0.56 \times 10^6\ \mathrm{Btu/yr})}$$

$$= \$36/10^6\ \mathrm{Btu{\cdot}yr} \qquad (5b)$$

Next, assuming a solar dehumidifier coefficient of performance COP_s of 0.5, the solar dehumidification cost, or solar cooling cost SCC, expressed as dollars per unit cooling capacity, will be

$$SCC = \frac{SHC}{COP_s}$$

$$= \frac{\$0.034}{0.5}$$

$$= \$0.068/10^6 \text{ J}\cdot\text{yr} \qquad (6a)$$

or

$$SCC = \frac{SHC}{COP_s}$$

$$= \frac{\$36}{0.5}$$

$$= \$72/10^6 \text{ Btu}\cdot\text{yr} \qquad (6b)$$

At 1980 fuel oil prices, the cost would be $\$5.54/1054 \times 10^6$ J ($\$5.54/10^6$ Btu) cooling capacity as shown in the following equation.

$$\frac{\$2.60}{1054 \times 10^6 \text{ J}}(2246.5 \times 10^6 \text{ J}) = \$5.54/1054 \times 10^6 \text{ J} \qquad (7a)$$

or

$$\frac{\$2.60}{10^6 \text{ Btu}}(2.13 \times 10^6 \text{ Btu}) = \$5.54/10^6 \text{ Btu} \qquad (7b)$$

On this basis, the simple payout period SPP, for the solar dehumidification system equals capital cost divided by savings, which can be expressed numerically by

$$SPP = \frac{\$0.68/10^6\ J \cdot yr}{\$5.54/(1054 \times 10^6\ J)}$$

$$= 13\ yr \qquad (8a)$$

or

$$SPP = \frac{\$72/10^6\ Btu \cdot yr}{\$5.54/10^6\ Btu}$$

$$= 13\ yr \qquad (8b)$$

However, in the case of electric-motor-driven chillers, the electrical energy required for generating chilled water in only 0.236 J/J (0.83 kWh/ton-h or 69.1 kWh/10^6 Btu) cooling capacity. At 1980 electric rates, the cost would be \$1.59/1054 x 10^6 J (\$1.59/10^6 Btu) cooling capacity. The simple payout period is then

$$SPP = \frac{\$0.068/10^6\ J \cdot yr}{\$1.59/(1054 \times 10^6\ J)}$$

$$= 45\ yr \qquad (9a)$$

or

$$SPP = \frac{\$72/10^6\ Btu \cdot yr}{\$1.59/10^6\ Btu}$$

$$= 45\ yr \qquad (9b)$$

The preceding payout periods are substantially higher than 5 years, which usually is considered a reasonable upper limit. Therefore, it is concluded that the solar regenerated-desiccant dehumidification system is not economically feasible at this time.

Boiler Stack Heat Recovery

The existing central heating and cooling plant contains five package-type water-tube boilers to supply steam for chiller turbines and building heating. The addition of stack-heat-recovery units to the boilers would improve the efficiency of the boilers by approximately 5 percent.

The heat-recovery units could be used to preheat combustion air or to preheat feed water. Preheating feed water as shown in figure 36 is the most practical application, providing equal energy savings with 25 percent less capital equipment cost. Feed water, currently heated to a temperature of 383 K (230° F), would be heated an additional 36.1 K (65° F) by the economizer.

In the existing plant configuration using steam-turbine-driven chillers, all five boilers would be modified. If electric-motor-driven chillers are installed, two boilers would be modified.

Energy analysis.- In the existing plant configuration, all five boilers would be modified. Annual energy savings would be 5 percent of the natural gas or number 2 fuel oil, or 0.05×1266 terajoules = 63.3 terajoules ($0.05 \times 1201 \times 10^9$ British thermal units = 60.1×10^9 British thermal units). In the electric-motor-driven chiller configuration, natural gas or fuel oil consumption would be reduced to 18 percent of the quantity used in the existing plant. Two boilers will provide adequate capacity for supplying building heat and intermittent operation of steam-turbine-driven chillers. Annual energy savings would be 5 percent of 225 terajoules (213×10^9 British thermal units), or 11.23 terajoules (10.65×10^9 British thermal units), with two boilers equipped with stack-heat-recovery units.

Cost analysis.- Capital cost estimates (tables XV and XVI) were made in end-of-year 1975 dollars. An escalation rate of 10 percent per year compounded to 1979 was used to determine capital cost invested when equipment becomes operational. Additional maintenance and operations costs, based on a man-year equivalent (MYE) of $28 000, were estimated as follows.

1. Five boilers with economizer; 0.5 MYE = $14 000

2. Two boilers with economizer; 0.25 MYE = $7000

These O&M costs are in 1976 dollars. The O&M costs shown in tables XVII and XVIII were calculated using a 10-percent compounded escalation rate to 1980 and a 5-percent compounded escalation rate from 1981 to 2000.

Alternative Fuels Evaluation

The purpose of this study was to consider alternative fuels to replace natural gas as the primary fuel in the JSC central heating and cooling plant. The three alternative fuels evaluated are lightweight oil, heavy oil, and coal.

Lightweight-oil system.- The existing boilers are all equipped with combination gas/oil burners that enable boiler operation on either natural gas or light fuel oil. These burners are relatively old and should be replaced if gas is not to be used as the primary fuel. The existing fuel oil system consists of a 379-kiloliter (100 000 gallon), ground-level fuel oil tank, fuel oil pumps, interconnecting piping, and controls, and is sized for standby use in the event of a temporary outage of the natural gas supply. Two additional 3180-kiloliter (840 000 gallon) light-oil storage tanks would be required to provide a 90-day supply of fuel oil for continuous operation with oil as the primary fuel. Delivery of oil to the plant could be by barge or direct pipeline; however; this report deals with barged oil only. A final project design should explore the possibility of using direct pipelines from a nearby refinery.

Heavy-oil system.- Conversion to heavy-weight oil will require removal of existing burners and the installation of new burners designed to burn the heavy oil. The existing light-oil storage tank can be used to store heavy oil; however, the addition of oil heaters will be required at the tank. Two additional 3180-kiloliter (840 000 gallon), ground-level storage tanks will be required to provide a 90-day fuel supply. Oil piping would be extended to the two additional oil storage tanks. Existing piping systems would require modification to accommodate the heavy oil, and new heavy-oil pumps would be required. An oil heater would be provided in the plant. Delivery of oil to the plant would be by barge or pipeline, similar to the arrangement for light oil. Firing of heavy oil at JSC also poses potential environmental problems.[2]

Coal-fired system.- Consideration was given to converting the existing gas- or oil-fired water-tube boilers to coal. It was determined, however, that it would not be feasible to convert this type boiler to coal firing because of the lack of space available for locating the equipment required to burn coal and remove ash. Also, the basic boiler design is not compatible with coal firing. Consideration was also given to replacing the existing gas- or oil-fired boilers in building 24 with coal-fired boilers located in that building. The available space in the building, however, was determined to be inadequate to house the required coal-burning equipment and ash hoppers.

It has been determined that the best and most feasible way to supply steam to the site using coal-burning equipment would be to build a complete new boiler plant adjacent to the existing building 24 plant. The existing gas- or oil-fired boiler plant would remain intact and could be operated on light oil or gas if in the event of a breakdown in the coal plant or a failure of the coal supply for any reason. The two plants would be connected by steam supply and return piping running in an underground tunnel between the two buildings.

The plant could consist of three 45 359-kg/hr (100 000 lb/hr) coal-fired boilers and accessory equipment housed in a building of similar construction to that of building 24. Boilers would be sufficient to supply the peak steam demand, and the third boiler would act as a standby.

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[2] Energy Outlook and Alternate Fuels Study - 1985-2000. Prepared by the Ralph M. Parsons Company for the NASA Johnson Space Center (job. no. 5494-1), November 1975.

Coal would be stored on the ground, in the open, behind the boiler plant. Open-site storage was chosen over enclosed storage because of the increased fire hazard with enclosed storage. The storage area would be enclosed by a low retaining wall. An electric-motor-driven coal scraper system could be used to scrape coal across the storage area and into an underground pit. An elevator would lift the coal to a coal bunker located at a higher elevation than the boilers. Coal would be supplied to the feederstokers of each boiler.

The two most feasible methods of transporting coal to the ground-level storage areas are by barge or rail. Barge shipments could be unloaded at the existing docks. Movement of coal from the docks to the storage area would require a conveyor; the conveyor should be an enclosed, overhead type. Coal would be discharged into the coalbin hopper below grade. Coal in the hopper would then be picked up by an elevator and discharged onto the coal storage pile at grade or discharged into the bunker for use in the boilers.

Delivery of coal by rail would require construction of a rail spur into the plant and purchase of land right-of-way along the route of the rail spur. Coal from the rail hopper cars would be dumped into an underground hopper, picked up by an elevator, and discharged to the outdoor storage or to the bunkers as required.

Coal-plant structure: The coal-plant building should be a single-story, flat-roof structure with precast exposed-aggregate-facing concrete wall panels. The building would be approximately 27.4 meters (90 feet) wide, 73.2 meters (240 feet) long, and 22.9 meters (75 feet) high. Mezzanine and second levels would be used for equipment at proper elevations above the ground floor.

Electrical system: Electrical service for the new boiler plant could be obtained by extending the existing 12.47-kilowatt feeders 2-5 and 1-5 from building 24. A selector switch should be provided to take service from either feeder. Two unit substations should be provided in lieu of a single substation to improve reliability and reduce fault current. Transformers should be 12 470-volt delta primary, 277/480-volt wye secondary with the neutral grounded at the transformer. Two motor control centers should be provided with a tie breaker to crossfeed in the event of a transformer failure. All electrical requirements for the project should be served out of the two motor control centers with conduit. Miscellaneous 120-volt power requirements should be provided by dry-type, 480-volt primary, 120/-volt secondary transformers.

Energy analysis.- The use of an alternate fuel would result in additional energy consumption because of the requirement for pumps, heaters, conveyors, and pollution control equipment. Because of the limited scope of this study, additional energy requirements were not calculated.

Cost analysis.- Capital costs for the primary alternate fuels are given in table XIX, and a summary of the detailed cost estimates is contained in a contractor report.[3] Total site energy costs are listed in table XX, which shows

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[3]Central Heating and Cooling Plant Concepts and Analysis. Prepared by Bernard Johnson, Inc., for the NASA Johnson Space Center (contract NAS9-14864), January 30, 1976.

fuel costs and additional O&M costs for each of the three alternate fuel systems in comparison with the existing system. The O&M costs were estimated as follows.

1. Number 2 fuel oil system; 2 MYE = $56 000

2. Number 6 fuel oil system; 4 MYE = $112 000

3. Coal system; 10 MYE = $280 000

These O&M costs are in 1976 dollars. The O&M costs shown in table XX were calculated using a 10-percent compounded escalation rate to 1980 and a 5-percent compounded escalation rate from 1981 to 2000. Table XX indicates that very favorable simple payback periods of approximately 3.5 years are obtained with either light or heavy oil. On the other hand, coal firing does not appear cost-effective at 11 years payback time. Firing of coal at JSC also poses potential environmental problems.[2]

CONCLUSIONS AND RECOMMENDATIONS

The conclusions resulting from this study are listed as follows.

1. Present cost and availability projections of energy (electricity and fuel) favor conversion to "all-electric" system concepts. Natural gas supply will not be terminated, but its cost will become prohibitive, reaching almost $0.106/m^3 ($3.00/10^3 ft^3) by 1980. Accordingly, conversion of at least three chillers to electric drive is justified and will save approximately 22 percent total energy compared to the baseline year. The simple payout period is 1.4 years, based on projected fuel and electricity prices.

2. In view of the potential variances in the relative cost of electricity and fuel in the future, it is recommended that a preliminary engineering report be accomplished for the dual-fuel engine-generator/electric-motor-driven chiller option. This option yields a 26-percent total energy savings and provides the flexibility of producing air-conditioning either by purchased electricity or by onsite-generated electricity, as dictated by the price of electricity and fuel. The engine generators could be added following the installation of the electric motors. Specific points to be investigated in the preliminary engineering report are as follows.

   a. Prime-mover selection (type and size of engine)

   b. Siting and installation requirements

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[2] Energy Outlook and Alternate Fuels Study - 1985-2000. Prepared by the Ralph M. Parsons Company for the NASA Johnson Space Center (job. no. 5494-1), November 1975.

c. Operating and maintenance costs, staffing requirements, and spare parts and overhaul facility requirements

d. Operational modes with respect to purchased power

3. Total energy concepts provide a minimum purchased electricity option and a total energy savings of 38 percent. In combination with solid-waste energy recovery and conversion from steam to a hot-water distribution system, as much as 46 percent total energy savings may be achieved. The capital cost of this option of approximately $20 million and the payback time approximately 15 years make it unattractive at presently projected fuel and electricity prices.

4. Incineration of JSC solid waste and recovery of its energy content is an economically viable option in combination with the existing system or with any of the other options investigated. Pyrolysis of the solid waste may be preferred in combination with onsite power generation (total energy or engine-generator/motor-driven chillers). In this case, the fuel gas produced by pyrolysis could be used in the prime mover, displacing either natural gas or deisel fuel.

5. No energy storage concepts that provide energy savings were identified; however, storage of chilled water using insulated tanks for peak shaving of air-conditioning loads appears attractive from an economic standpoint.

6. Use of solar energy as a supplemental heat source is not an economically feasible option at this time. The best application identified is domestic-hot-water heating, but investment paybacks are 30 years or more with hardware retrofits.

7. Recycling of JSC wastewater is technically feasible, but, at present and projected potable water prices, it is not economically attractive. The highest potential recycle load is 632 162 m^3/yr (167×10^6 gal/yr) for cooling-tower makeup.

8. Additional heating and air-conditioning energy savings may be obtained by further reducing rates of pretreated outside ventilation air in office buildings and high-bay areas.

9. Conversion of the central heating and cooling plant boilers from 100 percent natural gas firing to 100 percent fuel oil firing appears economically attractive at this time. Payout periods of approximately 3.5 years are obtained based on projected fuel prices. The plant is presently capable of firing light fuel oil for short durations.

10. Addition of boiler stack heat recovery to two of the five existing boilers would provide an annual total energy savings of 0.5 percent with a 6.7-year simple payout period. This option would be compatible with the electric-motor-driven chiller option discussed in item 1.

In view of the preceding conclusions, the following recommendations are submitted.

1. Implement the electric-drive chiller option.

2. Implement a program to reduce pretreated outside ventilation air, and initiate a design study of total enthalpy systems.

3. Implement the boiler stack heat recovery option.

4. Keep the engine-generator/electric-motor-driven chiller option open and periodically review it with respect to the cost of fuel.

5. Periodically review the option to incinerate solid waste with respect to the cost of fuel.

6. Perform further analyses of the energy storage system (chilled water) as applied to the electric-drive chiller option.

Lyndon B. Johnson Space Center
National Aeronautics and Space Administration
Houston, Texas, March 7, 1977
776-10-00-00-72

TABLE I.- SUMMARY OF ENERGY AND COST ANALYSIS

Concept	Energy Savings				Capital cost, dollars	Simple payback period, yr
	Percent	Electricity, MJ (kWh)	Oil, m^3 (gal)	Gas, m^3 (ft^3)		
Total energy, configuration 1	37.9	449.6 (124.9×10^6)	-39 747 (-10.5×10^6)	26.90×10^6 (950×10^6)	$25\ 090 \times 10^3$	13.0
With solid-waste pyrolysis, configuration 2	40.0	449.6 (124.9)	-39 747 (-10.5)	29.85 (1054)	25 400	11.0
With solid-waste pyrolysis, configuration 3	40.0	449.6 (124.9)	-37 854 (-10.0)	26.90 (950)	25 490	11.0
With 635-K (175° F) hot-water distribution, configuration 4	43.7	449.6 (124.9)	-39 747 (-10.5)	34.07 (1203)	28 628	10.6
With solid-waste pyrolysis and hot-water distribution, configuration 5	45.8	449.6 (124.9)	-37 854 (-10.0)	34.07 (1203)	29 028	10.2
Engine-generator/motor-driven chillers, configuration 1	24.0	—	-9 464 (-2.5)	26.90 (950)	3 982	2.0
With solid-waste incineration, configuration 2	26.1	—	-9 464 (-2.5)	28.46 (1005.0)	4 292	1.8
With solid-waste pyrolysis, configuration 3	26.1	—	-7 911 (-2.09)	26.90 (950.0)	4 382	1.8

TABLE I.- Concluded

Concept	Energy Savings				Capital cost, dollars	Simple payback period, yr
	Percent	Electricity, MJ (kWh)	Oil, m^3 (gal)	Gas, m^3 (ft^3)		
Electric-drive chillers (1)	22.6	-104.4 (-29.0×10^6)	--	26.90×10^6 (950×10^6)	$2\ 580 \times 10^3$	1.4
Solid-waste incineration	2.1	0	--	1.56 (55.0)	310	1.9
Solid-waste pyrolysis	2.1	0	--	1.56 (55.0)	400	2.5
Energy storage (1) (chilled water)	--	--	--	--	86	6.0
Steam turbine/absorption air-conditioning	<<1.0	--	--	--	(a)	(a)
Boiler stack heat recovery, 5 units	2.2	0	--	1.71 (60.5)	653	4.0
Boiler stack heat recovery, 2 units	.50	0	--	.30 (10.6)	224	6.7
Alternate fuels						
No. 2 fuel oil	<0	0	0	0	686	3.6
Heavy fuel oil	<0	0	0	0	1 782	3.5
Coal	<0	0	0	0	21 506	11.0

[a]No estimate made.

TABLE II.- DAILY ELECTRICAL LOAD PROFILES

Time	Electrical loads, MJ (kWh)					
	Design summer	Average winter	Average spring	Average summer	Average fall	Weekend/holiday
1 a.m.	52 200 (14 500)	39 960 (11 100)	40 680 (11 300)	45 360 (12 600)	42 480 (11 800)	39 600 (11 000)
2 a.m.	54 000 (15 000)	42 120 (11 700)	42 120 (11 700)	47 160 (13 100)	43 560 (12 100)	39 600 (11 000)
3 a.m.	54 000 (15 000)	42 120 (11 700)	42 120 (11 700)	47 160 (13 100)	43 560 (12 100)	39 600 (11 000)
4 a.m.	52 560 (14 600)	40 320 (11 200)	41 040 (11 400)	45 720 (12 700)	43 200 (12 000)	39 600 (11 000)
5 a.m.	54 000 (15 000)	42 120 (11 700)	42 120 (11 700)	47 160 (13 100)	43 200 (12 000)	39 600 (11 000)
6 a.m.	59 400 (16 500)	46 080 (12 800)	46 440 (12 900)	51 840 (14 400)	48 600 (13 500)	39 600 (11 000)
7 a.m.	71 640 (19 900)	55 800 (15 500)	55 800 (15 500)	62 640 (17 400)	58 320 (16 200)	43 200 (12 000)
8 a.m.	77 400 (21 500)	60 480 (16 800)	60 480 (16 800)	68 040 (18 900)	63 360 (17 600)	43 200 (12 000)
9 a.m.	77 760 (21 600)	61 200 (17 000)	60 480 (16 800)	68 400 (19 000)	63 360 (17 600)	43 200 (12 000)
10 a.m.	78 480 (21 800)	61 560 (17 100)	61 200 (17 000)	68 760 (19 100)	65 520 (18 200)	43 200 (12 000)
11 a.m.	78 480 (21 800)	62 640 (17 400)	61 200 (17 000)	68 760 (19 100)	65 160 (18 100)	43 200 (12 000)
12 m.	79 920 (22 200)	79 920 (22 200)	75 240 (20 900)	78 480 (21 800)	79 920 (22 200)	43 200 (12 000)
1 p.m.	79 920 (22 200)	79 920 (22 200)	75 240 (20 900)	78 480 (21 800)	79 920 (22 200)	43 200 (12 000)
2 p.m.	79 920 (22 200)	79 920 (22 200)	74 520 (20 700)	78 480 (21 800)	79 920 (22 200)	43 200 (12 000)
3 p.m.	78 480 (21 800)	62 640 (17 400)	61 200 (17 000)	69 480 (19 300)	66 600 (18 500)	43 200 (12 000)
4 p.m.	77 040 (21 400)	60 480 (16 800)	60 120 (16 700)	68 400 (19 000)	63 000 (17 500)	43 200 (12 000)
5 p.m.	69 120 (19 200)	54 000 (15 000)	54 000 (15 000)	60 480 (16 800)	56 520 (15 700)	43 200 (12 000)
6 p.m.	63 000 (17 500)	52 560 (14 600)	48 960 (13 600)	55 080 (15 300)	51 480 (14 300)	43 200 (12 000)
7 p.m.	61 200 (17 000)	49 320 (13 700)	47 520 (13 200)	53 640 (14 900)	50 040 (13 900)	39 600 (11 000)
8 p.m.	60 480 (16 800)	46 800 (13 000)	47 160 (13 100)	52 920 (14 700)	49 320 (13 700)	39 600 (11 000)
9 p.m.	59 400 (16 500)	46 080 (12 800)	46 440 (12 900)	51 840 (14 400)	48 960 (13 600)	39 600 (11 000)
10 p.m.	61 200 (17 000)	47 520 (13 200)	47 5[illegible] 13 200)	53 640 (14 900)	50 040 (13 900)	39 600 (11 000)
11 p.m.	60 480 (16 800)	47 160 (13 100)	47 1[illegible] (13 100)	52 920 (14 700)	49 680 (13 800)	39 600 (11 000)
12 p.m.	52 200 (14 500)	39 960 (11 100)	40 680 (11 300)	45 360 (12 600)	42 480 (11 300)	39 600 (11 000)

TABLE III.- MONTHLY ELECTRICAL LOADS

Date	Electrical load	
	Peak apparent power, kVA	Energy, MJ (kWh)
Feb. 1974	22 380	39 581 049 (10 994 736)
Mar. 1974	22 388	33 562 080 (9 322 800)
Apr. 1974	20 109	37 162 080 (10 322 800)
May 1974	20 288	35 093 001 (9 748 056)
June 1974	21 004	40 234 449 (11 176 236)
July 1974	22 083	38 006 841 (10 557 456)
Aug. 1974	22 200	40 934 764 (11 370 768)
Sept. 1974	21 838	41 787 144 (11 607 540)
Oct. 1974	22 328	34 916 400 (9 699 000)
Nov. 1974	24 689	36 765 219 (10 212 561)
Dec. 1974	26 975	35 121 758 (9 756 044)
Jan. 1975	29 595	34 577 323 (9 604 812)

TABLE IV.- JSC WATER DATA

Month, 1974	Monthly demands, m^3 (gal)						
	Potable water	Misc. cooling-tower and industrial losses	Bldg. 24 cooling-tower makeup	Bldg. 24 cooling-tower blowdown	Bldg. 24 boiler feed	Irrigation usage	Total wastewater
Jan.	62 989 (16 640×10^3)	1 601 (423×10^3)	42 851 (11 320×10^3)	8 971 (2 370×10^3)	2 279 (602×10^3)	--	25 230 (6 665×10^3)
Feb.	63 197 (16 695)	1 601 (423)	43 002 (11 360)	9 009 (2 380)	2 994 (791)	--	24 605 (6 500)
Mar.	68 683 (18 144)	1 601 (423)	46 712 (12 340)	9 766 (2 580)	2 972 (785)	--	27 164 (7 176)
Apr.	60 358 (15 945)	1 601 (423)	41 072 (10 850)	8 631 (2 280)	2 934 (775)	--	23 379 (6 176)
May	80 308 (21 215)	1 601 (423)	54 623 (14 430)	11 432 (3 020)	2 90[illegible] (768)	14 226 (3 758×10^3)	18 386 (4 857)
June	113 172 (29 897)	1 601 (423)	76 995 (20 340)	16 202 (4 280)	2 487 (657)	14 562 (3 847)	33 732 (8 911)
July	119 282 (31 511)	1 601 (423)	81 159 (21 440)	16 996 (4 490)	1 703 (450)	19 684 (5 200)	32 131 (8 488)
Aug.	88 147 (23 286)	1 601 (423)	59 923 (15 830)	12 530 (3 310)	1 003 (265)	19 752 (5 218)	18 397 (4 860)
Sept.	74 629 (19 715)	1 601 (423)	50 762 (13 4[illegible]0)	10 675 (2 820)	2 014 (532)	18 068 (4 773)	12 863 (3 398)
Oct.	80 451 (21 253)	1 601 (423)	54 813 (14 430)	11 470 (3 030)	2 093 (553)	--	33 410 (8 826)
Nov.	57 864 (15 286)	1 601 (423)	39 368 (10 400)	8 252 (2 180)	2 544 (672)	--	22 603 (5 971)
Dec.	60 154 (15 891)	1 601 (423)	40 882 (10 800)	8 555 (2 260)	2 461 (650)	--	23 746 (6 273)
Totals	929 234 (245 478)	19 212 (5076)	632 162 (167 000)	132 489 (35 000)	28 391 (7500)	86 292 (22 796)	295 646 (78 101)

TABLE V.- PROJECTED PRICES - BASELINE-CASE SCENARIO

[1975 prices]

Energy sources	1970	1975	1980	1985	1990	2000
Crude oil						
Cost, $\$/m^3$ ($/bbl)	20.00 (3.18)	55.98 (8.90)	81.77 (13.00)	100.64 (16.00)	125.80 (20.00)	176.11 (28.00)
Av cost increase per year, percent . . .	--	22.0	8.0	4.5	4.0	3.5
No. 2 fuel oil						
Cost, $\$/m^3$ ($/bbl)	21.39 (3.40)	76.61 (12.18)	94.35 (15.00)	110.07 (17.50)	138.38 (22.00)	188.69 (30.00)
Av cost increase per year, percent . . .	--	30.0	4.2	3.0	4.7	3.1
Residual fuel oil						
Cost, $\$/m^3$ ($/bbl)	23.40 (3.72)	76.74 (12.20)	97.49 (15.50)	106.93 (17.00)	132.09 (21.00)	182.40 (29.00)
Av cost increase per year, percent . . .	--	27.0	5.0	2.0	4.3	3.3
Natural gas						
Cost, $\$/m^3$ ($\$/10^3\ ft^3$)	0.008 (0.23)	0.053 (1.50)	0.104 (2.95)	0.134 (3.80)	0.159 (4.50)	0.194 (5.50)
Av cost increase per year, percent . . .	--	23.0	14.6	5.0	3.7	2.5
Electricity						
Cost, ¢/MJ (¢/kWh)	0.15 (0.54)	0.35 (1.26)	0.64 (2.30)	0.83 (3.00)	1.00 (3.60)	1.19 (4.30)
Av cost increase per year, percent . . .	--	23.0	8.0	4.5	3.8	2.00
Lignite						
Cost, $/Mg ($/ton)	2.15 (1.95)	5.90 (5.35)	11.79 (10.70)	14.77 (13.40)	18.30 (16.60)	24.80 (22.50)
Av cost increase per year, percent . . .	--	22.0	15.0	4.5	4.5	3.00
Bituminous coal						
Cost, $/Mg ($/ton)[a]	11.02 (10.00)	18.74 (17.00)	33.07 (30.00)	40.79 (37.00)	50.71 (46.00)	78.26 (71.00)
Av cost increase per year, percent . . .	--	11.0	12.0	4.5	4.5	3.00
Propane						
Cost, $\$/m^3$ (¢/gal)	15.32 (5.8)	75.29 (28.5)	121.52 (46.0)	195.49 (74.0)	287.95 (109.0)	515.14 (195.0)
Av cost increase per year, percent . . .	--	--	10.0	10.0	8.0	6.0
Butane						
Cost, $\$/m^3$ (¢/gal)	21.93 (8.3)	85.06 (32.2)	137.37 (52.0)	221.91 (84.0)	324.93 (123.0)	581.18 (220.0)
Av cost increase per year, percent . . .	--	--	10.0	10.0	8.0	6.0

[a]Includes transportation.

TABLE VI.- DELAVAL (ENTERPRISE) ENGINE PERFORMANCE DATA

[Model RV-16-4]

Power rating, kW	Percent rated load	sfc, J/J (Btu/kWh)	Heat recovery, MW (Btu/hr)		
			Jacket water at 353 K (175° F)	Lubrication oil at 353 K (175° F)	Exhaust at 394 K (250° F)
8000	110	2.49 (8499)	3.92 (13.4×10^6)	0.94 (3.2×10^6)	4.28 (14.6×10^6)
7274	100	2.47 (8445)	3.57 (12.2)	.85 (2.9)	3.92 (13.4)
6546	90	2.49 (8499)	3.22 (11.0)	.76 (2.6)	3.54 (12.1)
5820	80	2.50 (8533)	2.87 (9.8)	.67 (2.3)	3.13 (10.7)
5091	70	2.51 (8577)	2.49 (8.5)	.62 (2.1)	2.75 (9.4)

TABLE VII.- INITIAL OUTLAY FOR JSC TOTAL ENERGY CONCEPT, CONFIGURATION 1

Category	Cost in Jan. 1976
Engine-generators	
6 Delaval (Enterprise) dual-fuel prime-mover/generator sets, model RV-16-4, 7025 kW each set delivered in Houston at \$120/kW; mechanical installation at \$15/kW; electrical installation at \$15/kW; fuel oil, natural gas, steam, water, and lubrication oil piping and pumps at \$35/kW; controls at \$5/kW (\$190/kW×42 150 kW)	\$ 8 010 000
Air-cooled heat exchangers for jacket water and lubrication oil with capacity of 4.87 MW (16 624 000 Btu/hr) for each engine at \$110 000 each engine	660 000
6 exhaust-heat-recovery boilers with capacity of 4.28 MW (14 600 000 Btu/hr) recovered as 103.4-kN/m^2 (15 psig) steam at \$140 000 each	840 000
Subtotal	\$ 9 510 000
Electrical	
1 30-MVA step-up transormer, 3-phase, 600 V/12.4 kV	410 000
6 9000-KVA switchgear units at \$70 000 each	420 000
Subtotal	\$ 830 000
Air-conditioning	
Demolition	14 000
Modify 4 existing chillers	25 000
3 4843-kW (1377 ton) absorption chillers at \$34/kW (\$120/ton) installed	500 000
4 1678.5-kW (2250 hp) synchronous motors at \$153 000 each	612 000
7 0.30-m^3/sec (4800 gal/min) pumps, motors, and switchgear at \$46 600 each	326 000
7 cooling-tower modifications	57 000
Subtotal	\$ 1 534 000
Building - 51.2- by 29.6-m (168 by 97 ft) addition to the north end of bldg. 24 at \$538.20/m^2 (\$50/ft^2)	\$ 815 000
Miscellaneous - Interconnect plumbing, wiring, equipment and labor at 15 percent of above items	\$ 1 900 000
Subtotal (all components)	\$14 590 000
Engineering and site investigation at 8 percent of materials and labor	\$ 1 170 000
General contractor - Profit and overhead at 15 percent (no construction loan, no contingency) of materials and labor excluding engineering	\$ 2 190 000
Total turnkey - Nominal (confidence level, ±25 percent; Jan. 1976 dollars)	\$17 950 000
Total turnkey - Nominal mid-1979 dollars, assuming an inflation rate of 10 percent/yr compounded	\$25 090 000

TABLE VIII.- FAIRBANKS-MORSE ENGINE PERFORMANCE DATA

[Model 38 TDD8 1/8]

Power rating, kW	Percent rated load	sfc, J/J (Btu/kWh)	Heat recovery, MW (Btu/hr)		
			Jacket water at 394 K (250° F)	Lubrication oil at 355 K (180° F)	Exhaust at 394 K (250° F)
1707	110	2.73 (9 326)	0.38 (1.30×10^6)	0.56 (1.9×10^6)	1.02 (3.49×10^6)
1552	100	2.73 (9 326)	.35 (1.18)	.51 (1.73)	.93 (3.17)
1161	75	2.86 (9 764)	.28 (.95)	.41 (1.39)	.76 (2.58)
776	50	3.42 (11 675)	.24 (.82)	.35 (1.19)	.62 (2.10)

TABLE IX.- INITIAL COSTS FOR ENGINE-GENERATOR/ELECTRIC-DRIVE CHILLER OPTION, CONFIGURATION 1

Category	Cost in Jan. 1976
Engine-generators	
3 Fairbanks-Morse model 38 TDD8 1/8 dual-fuel engine-generator sets rated at 1611 kW each at \$130/kW delivered in Houston; mechanical and electrical installation at \$25/kW, fuel oil, natural gas, steam, water, and lubrication oil piping and pumps at \$30/kW, controls at \$10/kW (\$195/kW×4833 kW)	\$ 942 000
3 exhaust-heat-recovery boilers each with capacity of approximately 791 kW (2.7×10^6 Btu/hr) at 394.3 K (250° F), 103.4-kN/m^2 (15 psig) saturated steam at \$10.19/kW (\$100/hp) plus \$10.19/kW (\$100/hp) for interconnect plumbing, pumps, tanks, and other equipment (\$20.39/kW×2452 kW (\$200/hp×250 hp))	50 000
1 75 700-liter (20 000 gal) fuel oil storage tank complete with pumps and piping	11 000
3 2000-kVA, 480- to 600-V/4160-V transformers at \$24 333	73 000
3 2000-kVA switchgear	75 000
Modifications to 6 chillers at \$6333 each	38 000
Subtotal	\$1 189 000
Air-conditioning	
7 0.30-m^3/sec (4800 gal/min) pumps, motor switchgear, and plumbing at \$46 571 each	\$ 326 000
1 2321-kW (660 ton) absorption water chiller delivered and installed	81 000
7 cooling-tower pump and fan motor modifications at \$5714 each	40 000
3 1678.5-kW (2250 hp) synchronous motors geared to mate with existing chillers operating at 35 877 rad/min (5710 rpm) and 30 788 rad/min (4900 rpm) at \$133 333 each	400 000
Subtotal	\$ 847 000
Housing - 16.76 by 30.5 m at \$538.20/m^2 (55 by 100 ft at \$50/ft^2) and demolition at \$6000	\$ 281 000
Total - Materials and installation labor	\$2 317 000
Engineering at 8 percent of materials and labor	185 000
General contractor - Profit and overhead at 15 percent of materials and labor (no construction loan, no contingency), Jan. 1976 dollars	\$ 347 000
Total turnkey - Nominal (confidence level, ±30 percent; Jan. 1976 dollars)	\$2 849 000
Total turnkey - Mid-1979 dollars, assuming an escalation rate of 10 percent/yr compounded	\$3 982 000

TABLE X.- CAPITAL COSTS FOR ELECTRIC-MOTOR-DRIVEN CHILLERS OPTION

Category	Cost at end of 1975
Electrical (installed)	
3 1678.5-kW (2250 hp) electric motors with foundations, unit substations, transformers, and switches	$ 421 000
2 1200-A, 15-kV circuit breakers, and enclosure	111 000
914 m (3000 ft) of 3-phase, 15-kV, 600-A power cable in duct and manhole	116 000
1158 m (3800 ft) of 3-phase, 15-kV, 600-A armored cable	151 000
Mechanical (installed)	
3 speed-increaser gearboxes	122 000
Modifications to 6 chiller evaporators	28 000
7 circulation pumps, 0.30-m^3/sec (4800 gal/min) at 56 m (185 ft) with 186.5-kW (250 hp) electric motors	375 000
Modifications to cooling tower pumps and fans	65 000
1 7034-kW (2000 ton) electric-motor-driven chiller package including starter and switches	256 000
Demolition of 4 existing chiller unit foundations	8 000
Subtotal	$1 653 000
Engineering - At 8 percent of installed hardware cost	$ 132 240
General contractor - At 15 percent of installed hardware cost	$ 247 950
Total turnkey - Nominal (confidence level, ±10 percent)	$2 033 190
Total turnkey - Mid-1978 dollars, assuming escalation rate of 10 percent/yr compounded (2.5 yr at 10 percent = 1.269)	$2 580 118

TABLE XI.- ENERGY COST COMPARISON: ELECTRIC-MOTOR-DRIVEN CHILLER SYSTEM COMPARED TO EXISTING SYSTEM

Energy	Cost, dollars, for -					
	1979	1980	1985	1990	1995	2000
Steam-turbine-driven chiller system						
Natural gas	3150×10^{3}	3470×10^{3}	4620×10^{3}	5460×10^{3}	$6\ 070\times10^{3}$	$6\ 690\times10^{3}$
Electricity	2600	2850	3760	4490	4 920	5 360
Total	5750	6320	8380	9950	10 990	12 050
Electric-motor-driven chiller system						
Natural gas	670×10^{3}	740×10^{3}	980×10^{3}	$1\ 160\times10^{3}$	$1\ 290\times10^{3}$	$1\ 420\times10^{3}$
Electricity	3280	3590	4 740	5 660	6 200	6 750
Total[a]	3950	4330	5 720	6 820	7 490	8 170
Annual savings	1800	1990	2 660	3 130	3 500	3 880
Cumulative savings	1800	3790	15 930	30 610	47 370	65 980

[a]Payout period - $2 580 118 in 1.4 years.

TABLE XII.- INCINERATION WITH HEAT RECOVERY

COST SUMMARY

(a) Fuel costs

Category	Value at -			
	$1.19/GJ ($1.25/10^6 Btu)	$1.42/GJ ($1.50/10^6 Btu)	$1.90/GJ ($2.00/10^6 Btu)	$2.85/GJ ($3.00/10^6 Btu)
Incineration fuel yearly cost, dollars	7 475	8 970	11 960	17 940
Gross annual cost, dollars	125 475	126 970	129 960	135 940
Value of recovered heat, dollars	75 692	90 831	121 108	181 662
Net annual cost, dollars	49 783	36 139	8 852	-45 722
Annual savings, dollars	15 217	28 861	56 148	110 722
Simple payout period, yr	20.4	10.7	5.5	2.8

(b) System costs

Category	Cost, dollars
Total capital cost	310 000
Operating and maintenance cost per year	84 000
Yearly collection, hauling, and offsite disposal	34 000
Current contract cost yearly (as of Feb. 1975)	65 000

TABLE XIII.- PYROLYSIS COST SUMMARY

(a) Fuel costs

Category	Value at -			
	\$1.19/GJ (\$1.25/10^6 Btu)	\$1.42/GJ (\$1.50/10^6 Btu)	\$1.90/GJ (\$2.00/10^6 Btu)	\$2.85/GJ (\$3.00/10^6 Btu)
Utility fuel yearly cost, dollars	0	0	0	0
Gross annual cost, dollars	154 000	154 000	154 000	154 000
Value of gas produced, dollars	71 251	85 501	114 002	171 003
Net annual cost, dollars	82 749	68 499	39 998	-17 003
Annual savings, dollars	-17 749	-3 499	25 002	82 003
Simple payout period, yr	—	—	16.0	4.9

(b) System costs

Category	Cost, dollars
Total capital cost	400 000
O&M cost per year	120 000
Yearly collection, hauling, and offsite disposal	34 000
Current contract cost yearly (as of Feb. 1975)	65 000

TABLE XIV.- CHARACTERISTICS OF DESICCANT SYSTEMS

System	Size	Status	Operating temperature,[a] K (°F)	COP	Heat rejection	Drying regime	
						Temperature	Humidity
Liquid absorbent	Commercial	Available	333 to 353 (140 to 175)	[b]0.5	Cooling tower	Low	High
Rotary - sieve	Commercial	Available	353 to 394 (175 to 250)	—	Regenerative	High	Low
	Residential	Development (2 to 5 yr)	[c]394 (250)	[b].7	Evaporative	High	Low
Rotary - gel	Commercial	Available	353 to 394 (175 to 350)	.8	Regenerative	Moderate	Moderate
	Residential	Development (1 yr)	—	—	Regenerative	Moderate	Moderate
Rotary - lithium chloride	Commercial	Available	339 to 394 (150 to 250)	—	Regenerative	High	Low
Bed - gel	Commercial	Available	339 to 394 (150 to 250)	—	Regenerative	Moderate	Moderate

[a]Low temperatures are estimated values; high temperatures are demonstrated.

[b]Based on test.

[c]Design point temperature for 80 percent solar.

TABLE XV.- INITIAL COSTS OF BOILER STACK-HEAT-RECOVERY OPTION (5 UNITS)

Category	Cost at end of 1975
Hardware (installed)	
5 economizers at \$64 000 each (includes blowers, structural supports, and stack modifications)	\$320 000
Feed-water piping - at \$1200 each (includes insulation and valves)	6 000
Subtotal	\$326 000
Engineering - At 8 percent of installed hardware cost	\$ 26 080
General contractor - At 15 percent of installed hardware cost	\$ 48 900
Total turnkey - Nominal (confidence level, ±10 percent; \$360 882 to \$441 078)	\$400 980
Total turnkey - Mid-1979 dollars, assuming escalation rate of 10 percent/yr compounded (3.5 yr at 10 percent = 1.3975)	\$560 370

TABLE XVI.- INITIAL COSTS OF BOILER STACK-HEAT-RECOVERY OPTION (2 UNITS)

Category	Cost at end of 1975
Hardware (installed)	
2 economizers at $64 000 each (includes blowers, structural supports, and stack modifications)	$128 000
Feed-water piping - at $1200 each	2 400
Subtotal	$130 400
Engineering - At 8 percent of installed hardware cost	$ 10 432
General contractor - At 15 percent of installed hardware cost	$ 19 560
Total turnkey - Nominal (confidence level, ±10 percent; $144 353 to $176 431)	$160 392
Total turnkey - Mid-1979 dollars, assuming escalation rate of 10 percent/yr compounded (3.5 yr at 10 percent = 1.3975)	$224 148

TABLE XVII.- ENERGY COST COMPARISON: BOILER STACK-HEAT-RECOVERY UNITS ON FIVE BOILERS

COMPARED TO EXISTING SYSTEMS

Energy	Cost, dollars, for -					
	1979	1980	1985	1990	1995	2000
Existing system						
Natural gas	3150×10^3	3470×10^3	4620×10^3	5460×10^3	$6\ 070\times10^3$	$6\ 690\times10^3$
Electricity	2600	2850	3760	4490	4 920	5 360
Total	5750	6320	8380	9950	10 990	12 050
System with boiler stacks heat recovery						
Natural gas	2990×10^3	3300×10^3	4390×10^3	5180×10^3	$5\ 770\times10^3$	$6\ 350\times10^3$
Electricity	2600	2850	3760	4490	4 920	5 360
Additional O&M cost	19	20	26	33	42	54
Total[a]	5609	6170	8176	9703	10 732	11 764
Annual savings	141	150	204	247	258	286
Cumulative savings	141	290	1240	2380	3 650	5 030

[a]Payout period — $653 192 in approximately 4 years.

TABLE XVIII.- ENERGY COST COMPARISON: ELECTRIC-DRIVE CHILLERS WITH BOILER STACK HEAT RECOVERY COMPARED TO ELECTRIC-DRIVE CHILLERS WITHOUT STACK HEAT RECOVERY

Energy	Cost, dollars, for -					
	1979	1980	1985	1990	1995	2000
Electric-drive chillers with no economizers						
Natural gas	670×10^3	740×10^3	980×10^3	1160×10^3	1290×10^3	1420×10^3
Electricity	3280	3590	4740	5660	6200	6750
Total	3950	4330	5720	6820	7490	8170
Electric-drive chillers and two boilers with economizers						
Natural gas	630×10^3	700×10^3	930×10^3	1100×10^3	1230×10^3	1350×10^3
Electricity	3280	3590	4740	5660	6200	6750
Additional O&M cost	9	10	13	17	21	27
Total[a]	3919	4300	5683	6777	7451	8127
Annual savings	31	30	37	43	39	43
Cumulative savings	31	61	240	450	660	880

[a]Payout period — $224 148 in approximately 6.7 years.

TABLE XIX.- ALTERNATE FUEL SYSTEMS

CAPITAL COST SUMMARY

Fuel system	1979 cost, dollars
No. 2 fuel oil system	686 000
No. 6 fuel oil system	1 782 000
Coal system	21 506 000

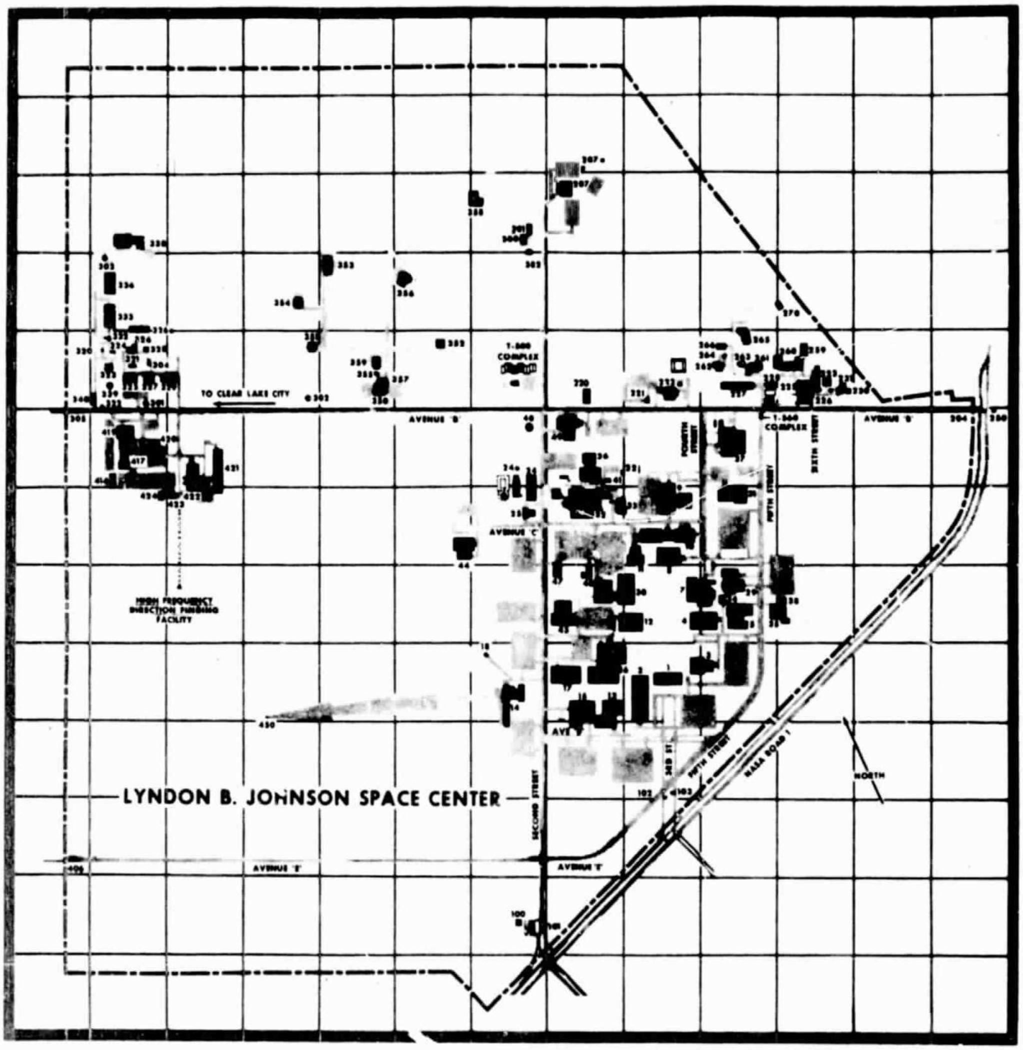
TABLE XX.- ENERGY COST COMPARISON: EXISTING NATURAL GAS SYSTEM COMPARED TO THREE ALTERNATIVE FUEL SYSTEMS

Energy	Cost, dollars, for -					
	1979	1980	1985	1990	1995	2000
Existing natural gas system						
Natural gas	3150×10³	3470×10³	4 620×10³	5 460×10³	6 070×10³	6 690×10³
Electricity	2600	2850	3 760	4 490	4 920	5 360
Total	5750	6320	8 380	9 950	10 990	12 050
No. 2 fuel oil system						
No. 2 fuel oil	3000×10³	3120×10³	3 600×10³	4 560×10³	5 340×10³	6 170×10³
Natural gas	130	150	200	230	260	290
Electricity	2600	2850	3 760	4 490	4 920	5 360
Additional O&M	74	82	105	133	170	217
Total[a]	5804	6202	7 665	9 413	10 690	12 037
Annual savings	-54	118	715	537	300	13
Cumulative savings	-54	70	274	577	776	834
No. 6 oil system						
No. 6 fuel oil	2520×10³	2690×10³	3 310×10³	4 140×10³	4 970×10³	5 800×10³
Natural gas	130	150	200	230	260	290
Electricity	2600	2850	3 760	4 490	4 920	5 360
Additional O&M	150	160	200	260	330	410
Total[b]	5400	5850	7 470	9 120	10 480	11 860
Annual savings	350	470	910	830	510	190
Cumulative savings	350	820	4 730	8 990	12 180	13 760
Coal system						
Coal	1300×10³	1410×10³	1 750×10³	2 170×10³	2 760×10³	3 360×10³
Natural gas	130	150	200	230	260	290
Electricity	2600	2850	3 760	4 490	4 920	5 360
Additional O&M	370	410	520	670	850	1 090
Total[c]	4400	4820	6 230	7 560	8 790	10 100
Annual savings	1350	1500	2 150	2 390	2 200	1 950
Cumulative savings	1350	2850	12 510	23 940	35 290	45 540

[a]Payout period — $686 000 in approximately 3.6 years.

[b]Payout period — $1 782 000 in approximately 3.5 years.

[c]Payout period — $21 506 000 in approximately 11 years.



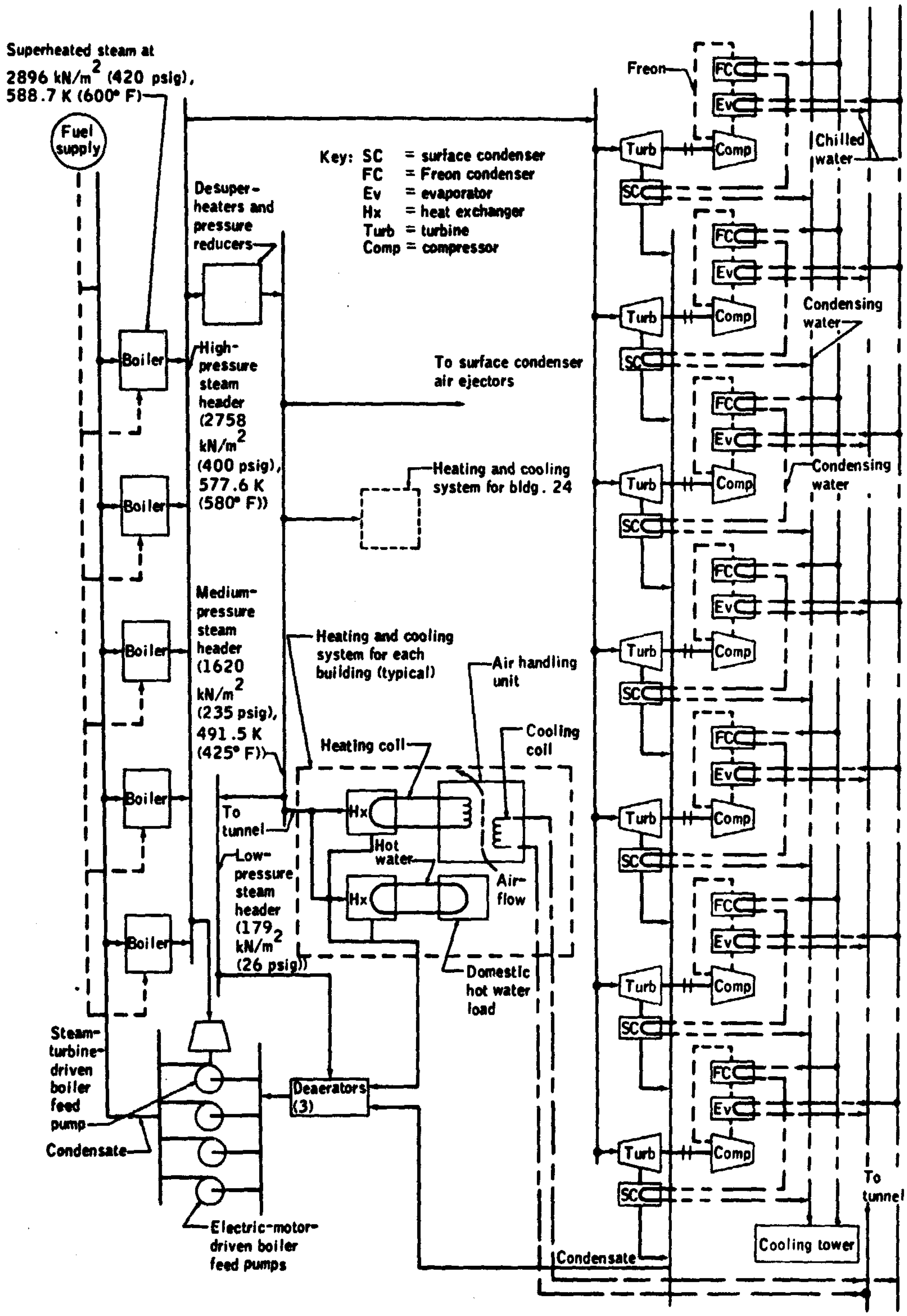
FACILITY NO.	FACILITY TITLE	LOCATION
1	PROJECT MANAGEMENT BUILDING	I-8
2	AUDITORIUM AND PUBLIC AFFAIRS FACILITY	I-8
3	CENTRAL CAFETERIA	I-8, 10
4	FLIGHT OPERATIONS FACILITY	H-8, 10
5	MISSION SIMULATION AND TRAINING FACILITY	H-10
7	LIFE SYSTEMS LABORATORY	H-8, 10
8	PHOTOGRAPHIC TECHNOLOGY LABORATORY	G-8
9	TECHNICAL SERVICES FACILITY	G-8
10	TECHNICAL SERVICES SHOP	G-8
11	BRANCH CAFETERIA	G-8
12	CENTRAL DATA OFFICE	H-8
13	STRUCTURES AND MECHANICS LABORATORY	I-8
14	ANECHOIC CHAMBER TEST FACILITY	I-7
15	INSTRUMENTATION AND ELECTRONIC SYSTEMS LABORATORY	I-8
16	SPACECRAFT SYSTEMS LABORATORY	I-8
17	EARTH RESOURCES APPLICATIONS DEVELOPMENT LABORATORY	I-8
18	RADAR BORESIGHT RANGE CONTROL BUILDING	I-7
24	CENTRAL HEATING AND COOLING PLANT	F, G-7
25	FIRE STATION	G-7
29	FLIGHT ACCELERATION FACILITY	H-10
30	MISSION CONTROL CENTER - HOUSTON	H-8, 9
31	PLANETARY AND EARTH SCIENCES LABORATORY	G-10
32	SPACE ENVIRONMENT SIMULATION LABORATORY	G-8
33	SPACE ENVIRONMENT EFFECTS LABORATORY	G-8
34	FLIGHT ACCELERATION MOTOR GENERATOR BUILDING	H-10
35	FLIGHT CREW TRAINING FACILITY	H-10, 11
36	BIOENGINEERING AND TEST SUPPORT FACILITY	F-8
37	LIFE SCIENCES LABORATORY	F-10
38	WATER IMMERSION MOCKUPS STORAGE FACILITY	H-11
40	ELEVATED WATER STORAGE TANK	F-7
41	HYPO/HYPERBARIC TRAINING FACILITY	F-8
44	ELECTRONIC SYSTEMS COMPATIBILITY FACILITY	G-8
45	PROJECT ENGINEERING OFFICES AND TECHNICAL LIBRARY	H-8
47	SOUTHWESTERN BELL TELEPHONE COMPANY FACILITY	H-8
48	EMERGENCY POWER BUILDING (MCC-H)	H-8
49	VIBRATION AND ACOUSTIC TEST FACILITY	F-8

FACILITY NO.	FACILITY TITLE	LOCATION
100	SECURITY CONTROL CENTER	L-7
101	SECOND STREET GATEHOUSE	L-7
102	THIRD STREET EXIT GATEHOUSE	J-8
103	THIRD STREET ENTRY GATEHOUSE	J-8
204	AVENUE B EAST GATEHOUSE	F-13
207	ROBERT R. GILRUTH RECREATION FACILITY	C-8
220	TEST ARTICLE PREPARATION FACILITY	E-8
221	138 KV ELECTRICAL SUBSTATION	E-8
222	ATMOSPHERIC RE-ENTRY MATERIALS AND STRUCTURES EVALUATION FACILITY	E-8
223	SEWAGE TREATMENT PLANT	E-11
225	ADMINISTRATION SUPPORT FACILITY ANNEX	E-11
226	ADMINISTRATION SUPPORT FACILITY	E-11
227	PRINTING AND REPRODUCTION FACILITY	E-10
228	RECOVERY QUARANTINE EQUIPMENT STORAGE FACILITY	E-10
230	UTILITY ANNEX	E-11
231	CUSTODIAL STORAGE BUILDING	E-11
250	BARGE DOCKING FACILITY	F-13
258	SPACECRAFT RECOVERY EQUIPMENT STORAGE BUILDING	E-11
260	TRAINING AND TEST FACILITY	E-11
261	PLANETARY AND EARTH SCIENCES LABORATORY ANNEX A	E-10
262	ARC JET, RADIANT HEATING AND ACOUSTIC TEST FACILITY	E-10
263	HEALTH PHYSICS LABORATORY	E-10
264	BIOLOGICAL INCINERATOR	E-10
265	PLANETARY AND EARTH SCIENCES LABORATORY ANNEX B	E-10
266	BIOMEDICAL RESEARCH SUPPORT FACILITY	E-10
270	SOLAR TELESCOPE FACILITY	D-11
301	WATER WELL NO. 1	E-2
302	WATER WELL NO. 2	E-4
303	WATER WELL NO. 3	D-2
304	MAINTENANCE PAINTING FACILITY	E-2
305	AVENUE B WEST GATEHOUSE	F-1
320	STEAM CLEANING FACILITY	E-2
321	MAINTENANCE EQUIPMENT FACILITY	E-2
322	WATER TREATMENT PLANT BUILDING	E-2
323	CONSTRUCTION WORK CONTROL CENTER	E-2

FACILITY NO.	FACILITY TITLE	LOCATION
324	CLASSIFIED WASTE DISINTEGRATOR FACILITY	E-2
325	MAINTENANCE MATERIALS STAGING AND SHOP FACILITY	E-2
326	GROUNDS EQUIPMENT MAINTENANCE FACILITY	E-2
327	MAINTENANCE WORK CONTROL CENTER	E-2
328	CONSTRUCTION MATERIALS STAGING FACILITY	E-2
329	MAINTENANCE SHOP FACILITY	E-2, 3
332	CENTRAL WASTE COLLECTION FACILITY	E-2
333	TEMPORARY STORAGE WAREHOUSE	D-2
335	TEST EQUIPMENT WAREHOUSE	D-2
336	WATER/LAND IMPACT TEST FACILITY	C-2
339	ABOVE GROUND WATER STORAGE TANK	E-2
340	GAS METERING STATION	E-2
350	THERMOCHEMICAL TEST AREA SUPPORT LABORATORY	E-6
351	THERMAL VACUUM TEST FACILITY	E-4
352	PYROTECHNICS TEST FACILITY	E-6
353	PROPULSION TEST FACILITY	D-6
354	CRYOGENICS TEST FACILITY	D-4
355	CHEMICAL STORAGE BUILDING	E-6
356	FLUID SYSTEMS TEST FACILITY	D-6
357	THERMOCHEMICAL TEST AREA GATEHOUSE	E-6
358	PROPELLANT FUEL WASTE PRETREATMENT FACILITY	C-7
359	THERMOCHEMICAL EQUIPMENT STORAGE FACILITY	E-6
360	HAZARDOUS MATERIAL STORAGE FACILITY	C-7
361	CENTRAL GAS CYLINDER STORAGE SHED	C-7
362	DRUM STORAGE SHED	C-7
400	AVENUE E GATEHOUSE	K-1
416	INSTALLATION SUPPORT FACILITY	F-2
417	GARAGE	F-2
419	LOGISTICAL SERVICE OFFICES	F-2
420	SHIPPING AND RECEIVING WAREHOUSE	F-2
421	GENERAL SUPPLY WAREHOUSE	F-3
422	LOGISTIC SUPPORT WAREHOUSE	F-3
423	CENTER SUPPORT FACILITY	G-3
424	INTERIM FILM REPOSITORY FACILITY	E-2
450	MICROWAVE TRANSMITTING ANTENNA TOWER	I-4

(a) Overview of total facilities.

Figure 1.- Johnson Space Center facilities.



(b) Schematic of Central Heating and Cooling Plant (Building 24).

Figure 1.- Concluded.

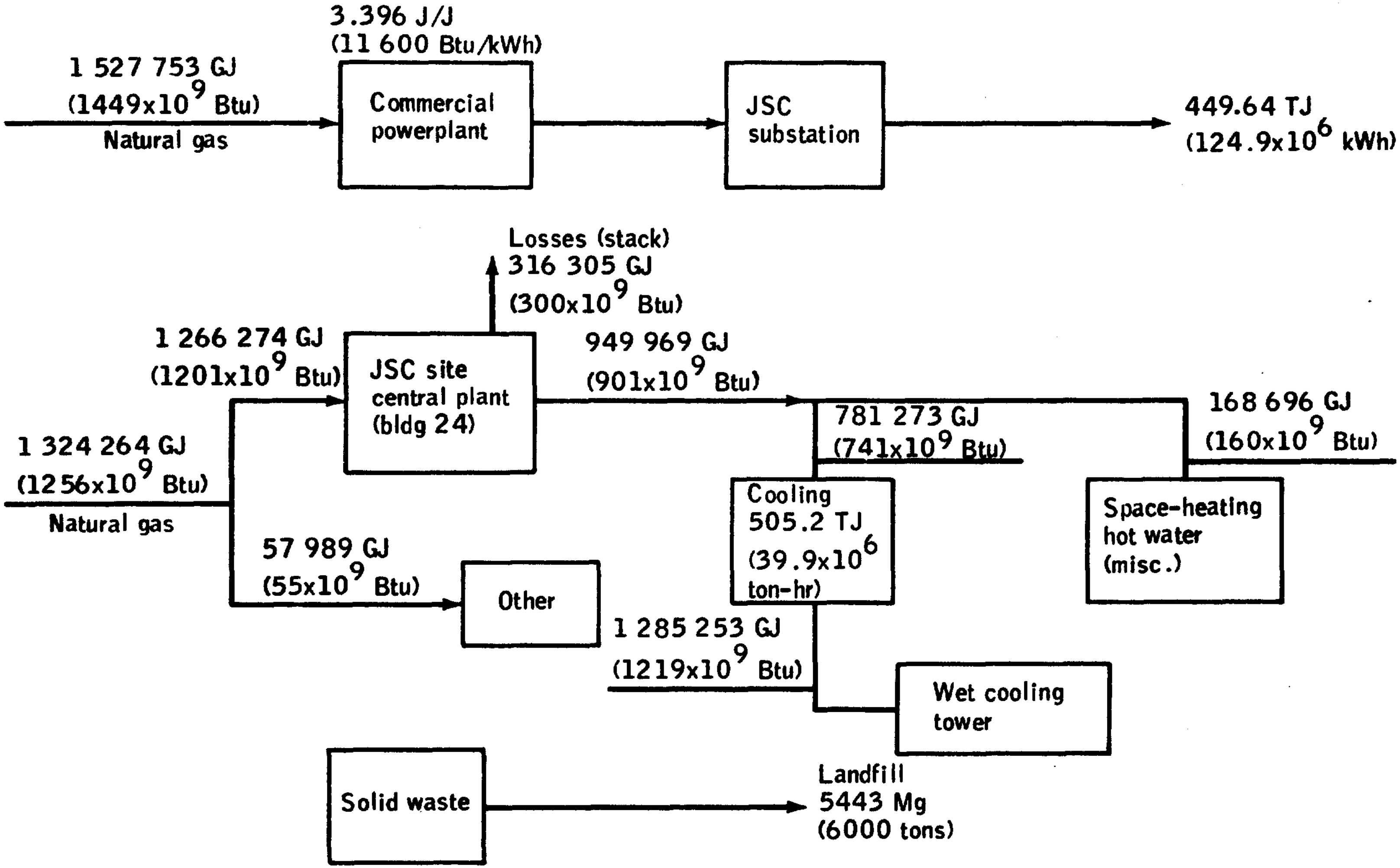
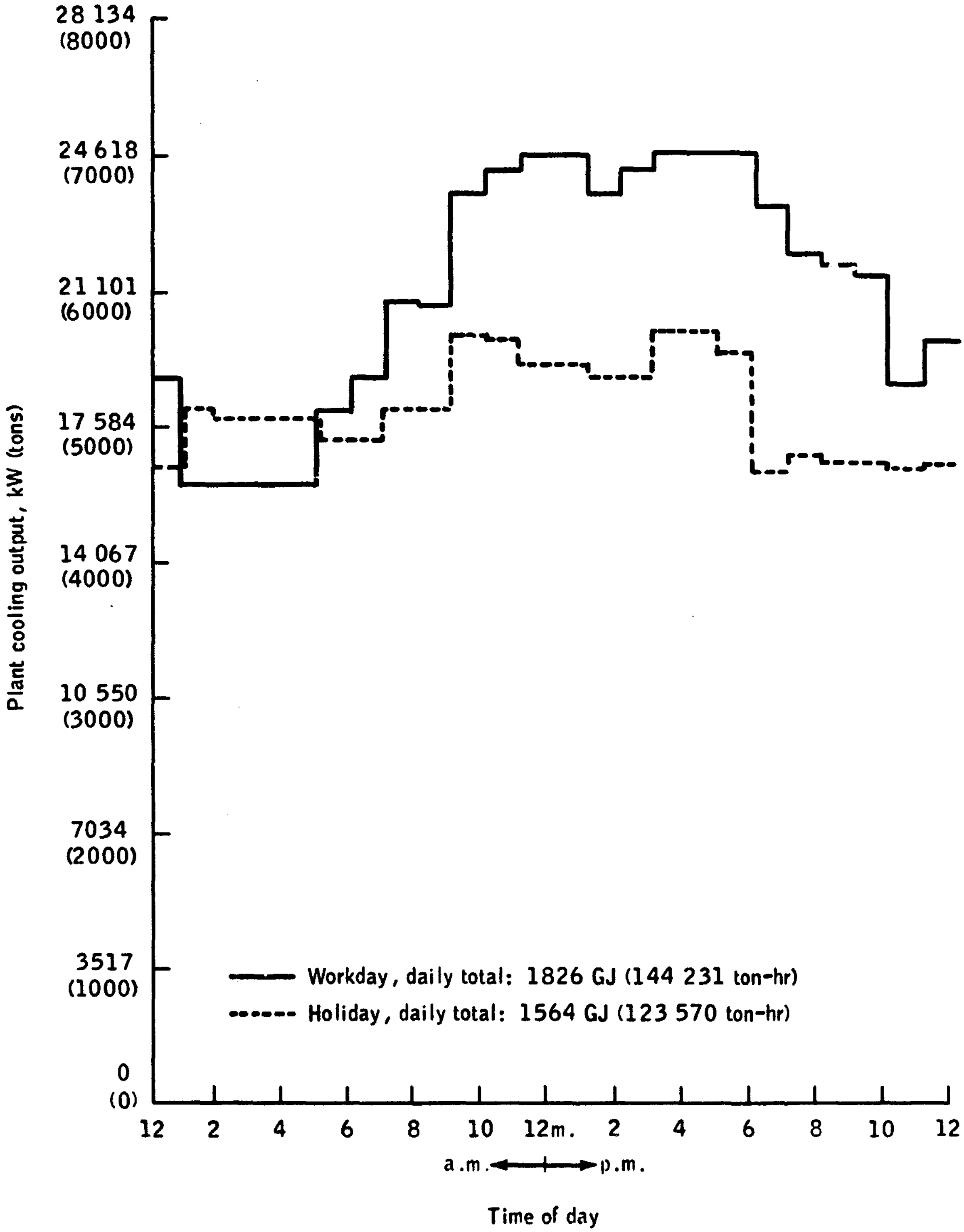
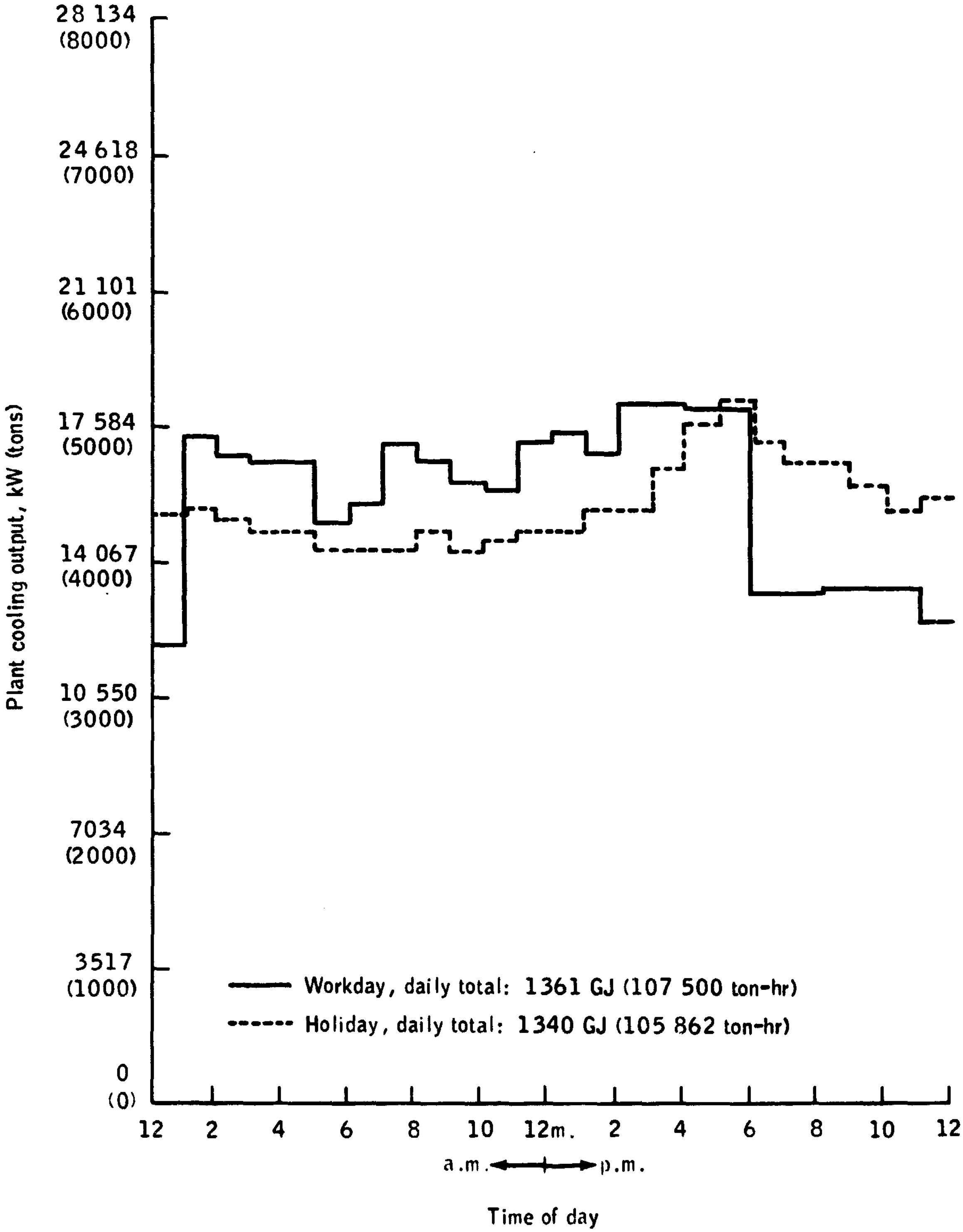


Figure 2.- Annual JSC energy consumption (February 1974 to January 1975).



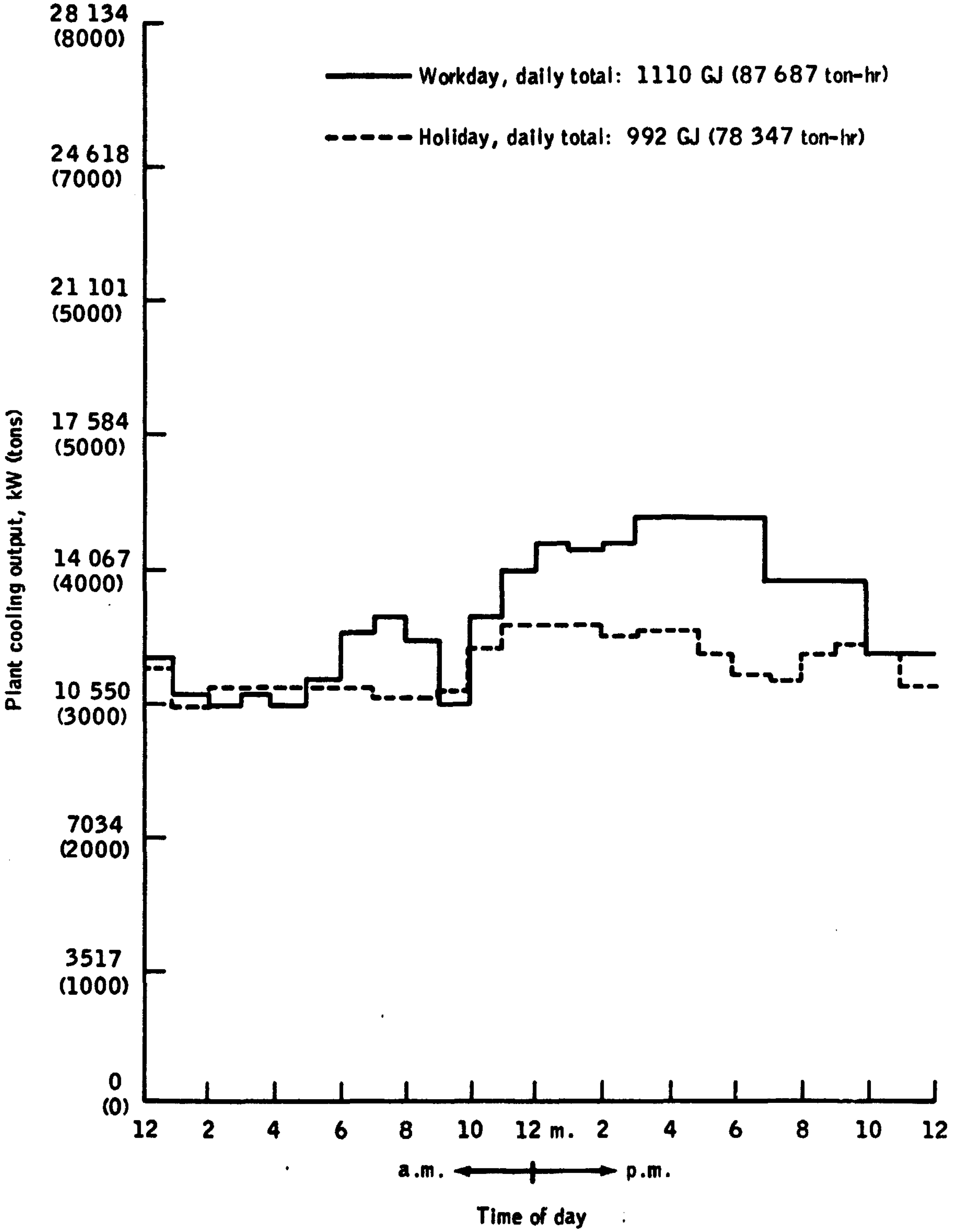
(a) Summer.

Figure 3.- Maximum, minimum, and seasonal JSC cooling loads.



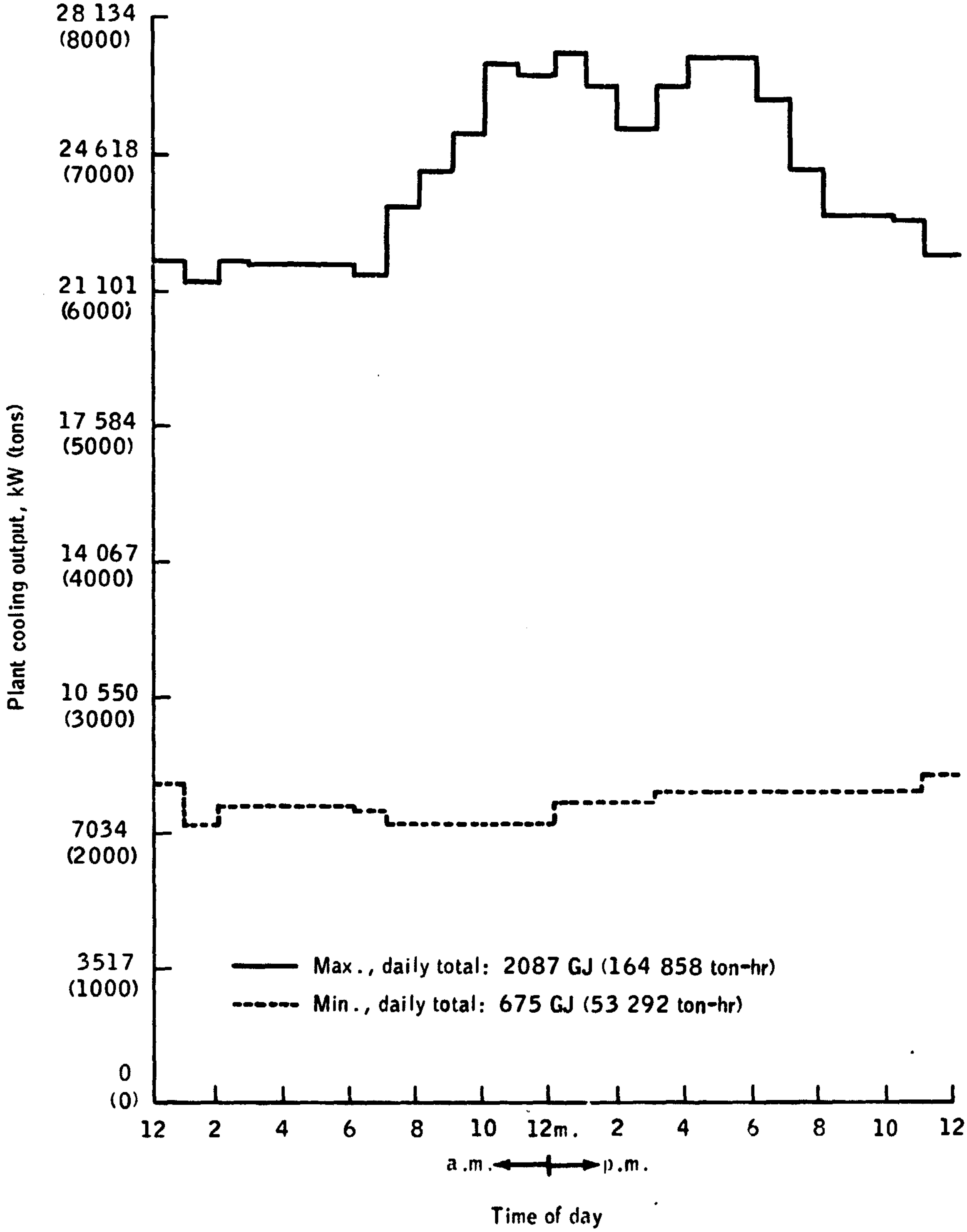
(b) Fall.

Figure 3.- Continued.



(c) Winter.

Figure 3.- Continued.



(d) Extremes.

Figure 3.- Concluded.

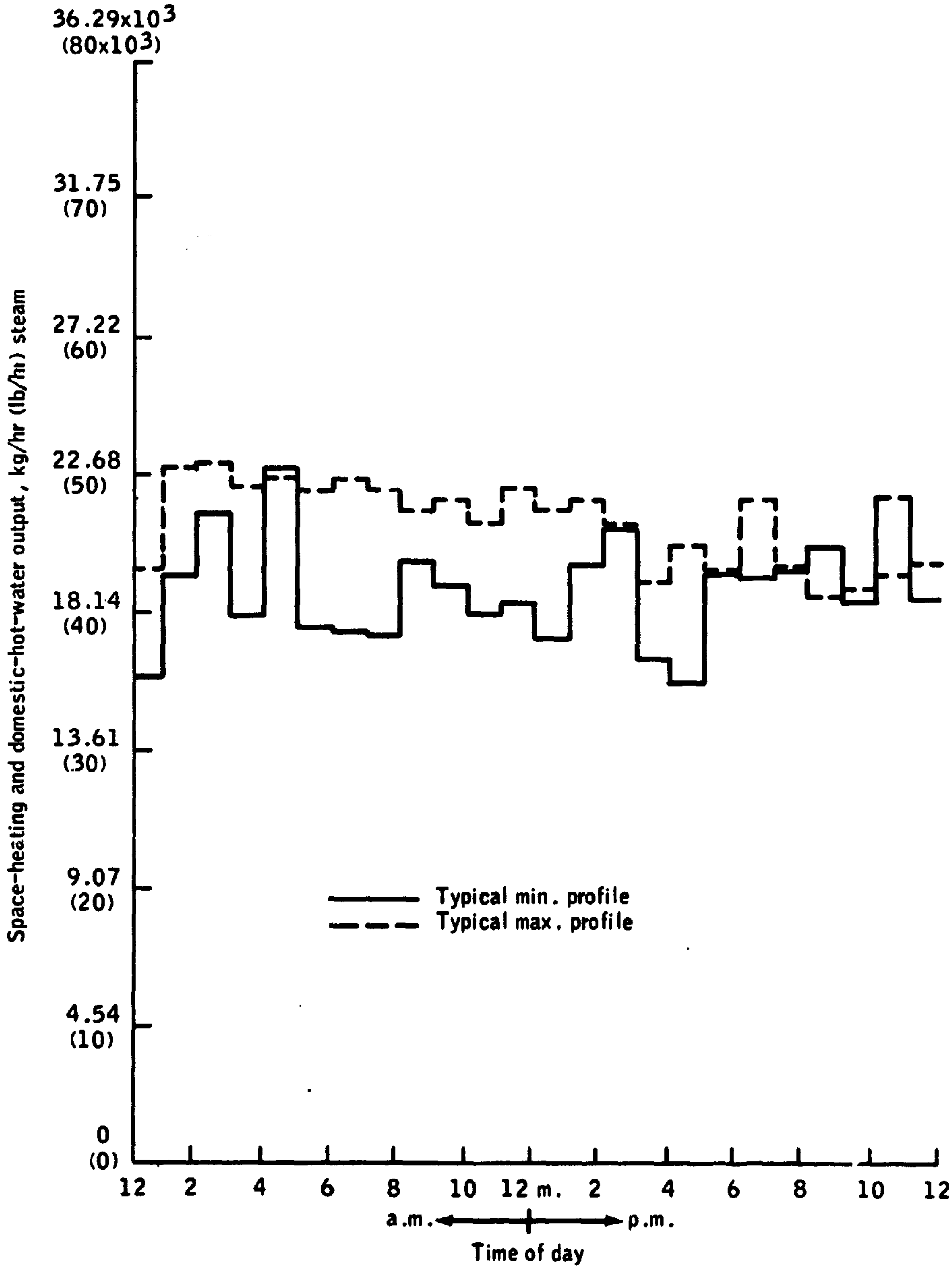


Figure 4.- Maximum and minimum JSC heating loads.

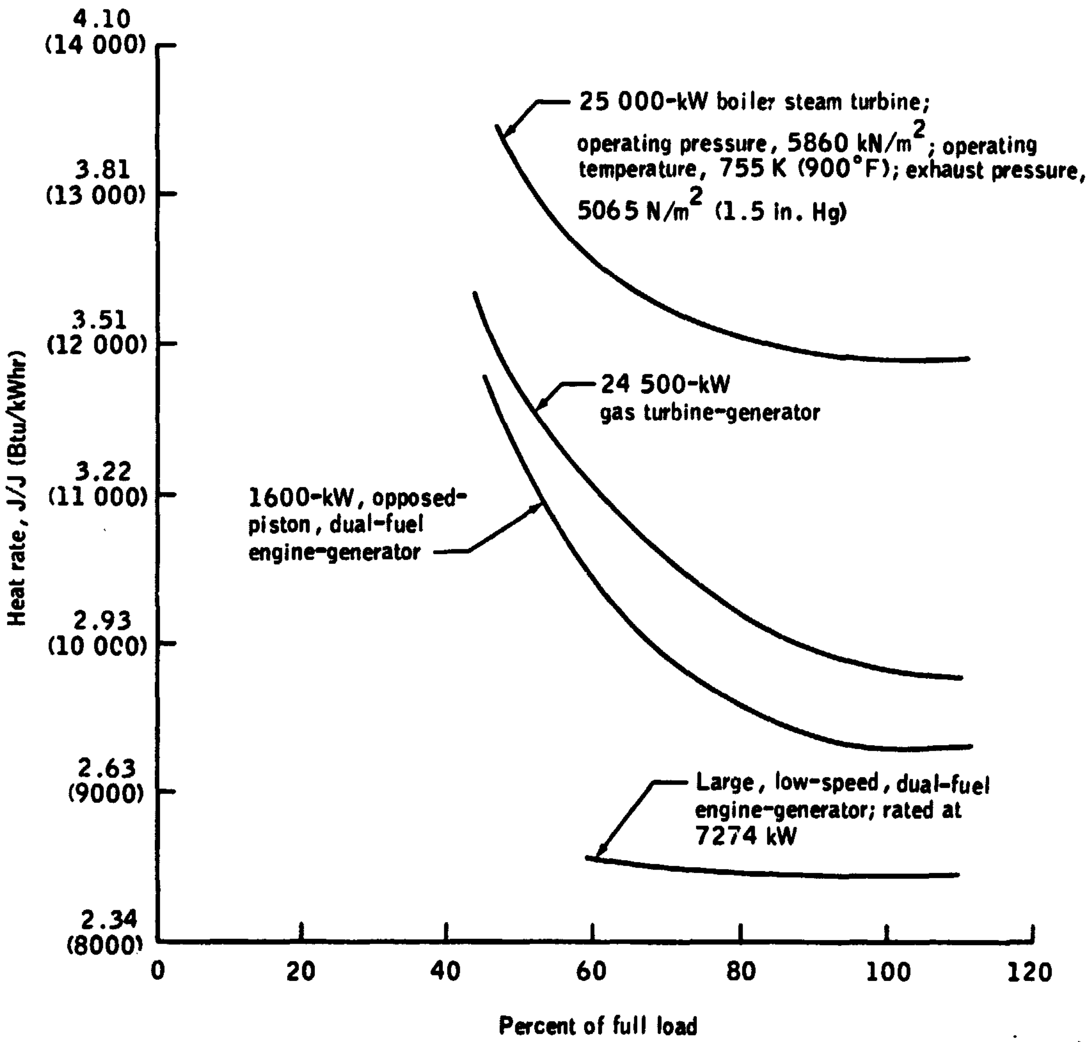


Figure 5.- Typical heat rates of steam turbine, gas turbine, and dual-fuel (reciprocating engine) electrical powerplants.

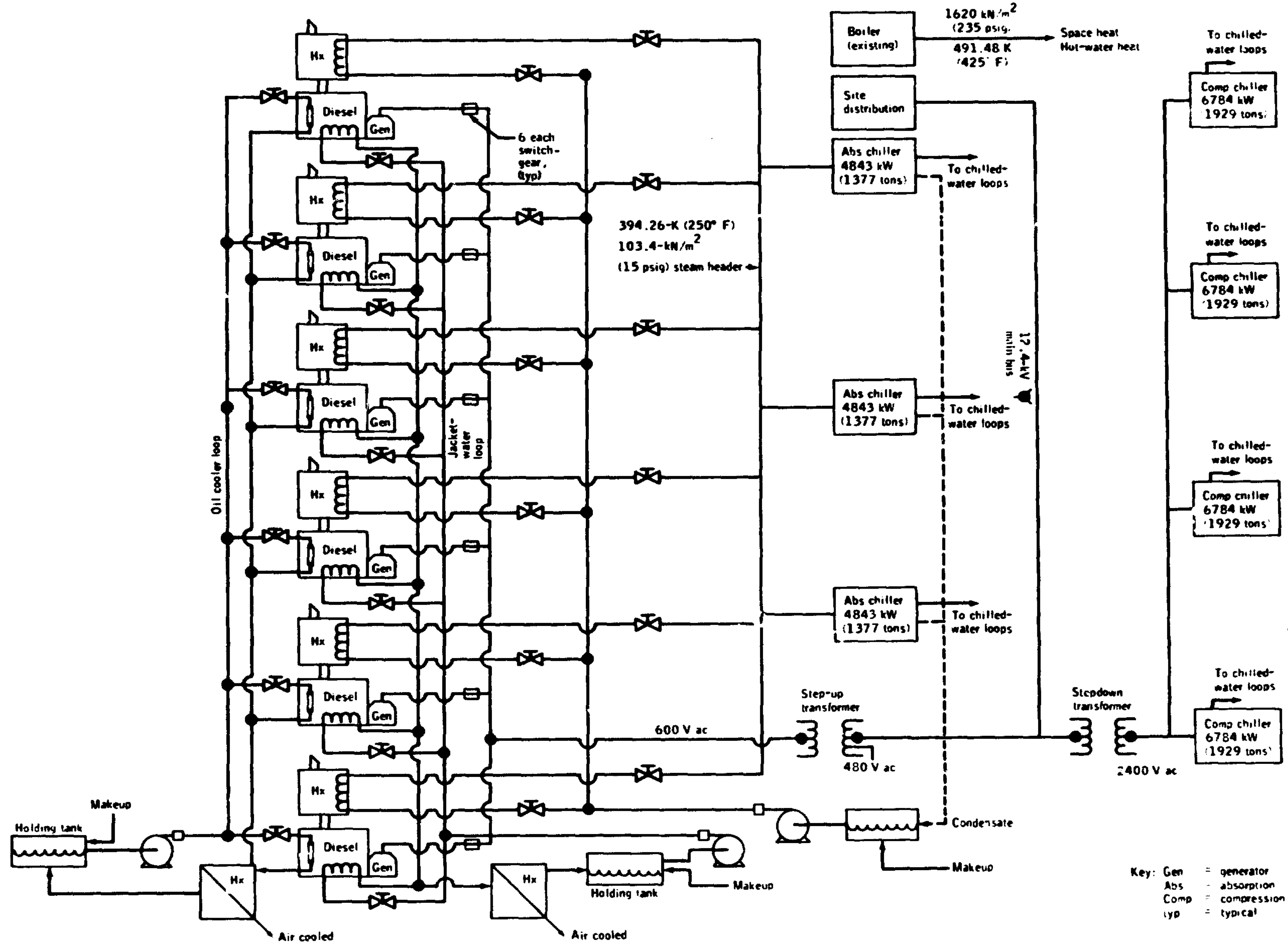


Figure 6.- The JSC total energy system schematic.

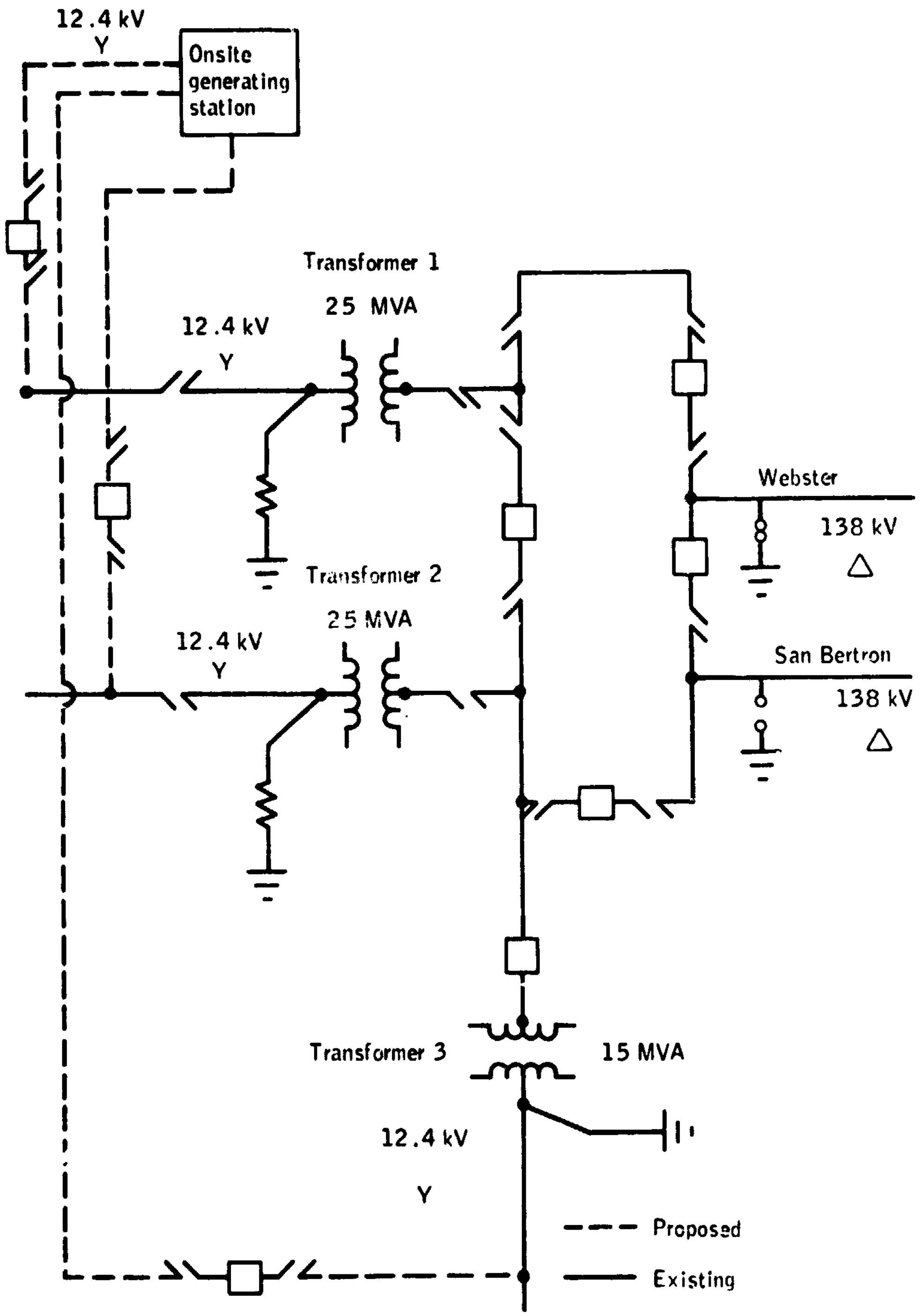


Figure 7.- Alternate onsite power generation for facility number 221 (138-kV electric substation).

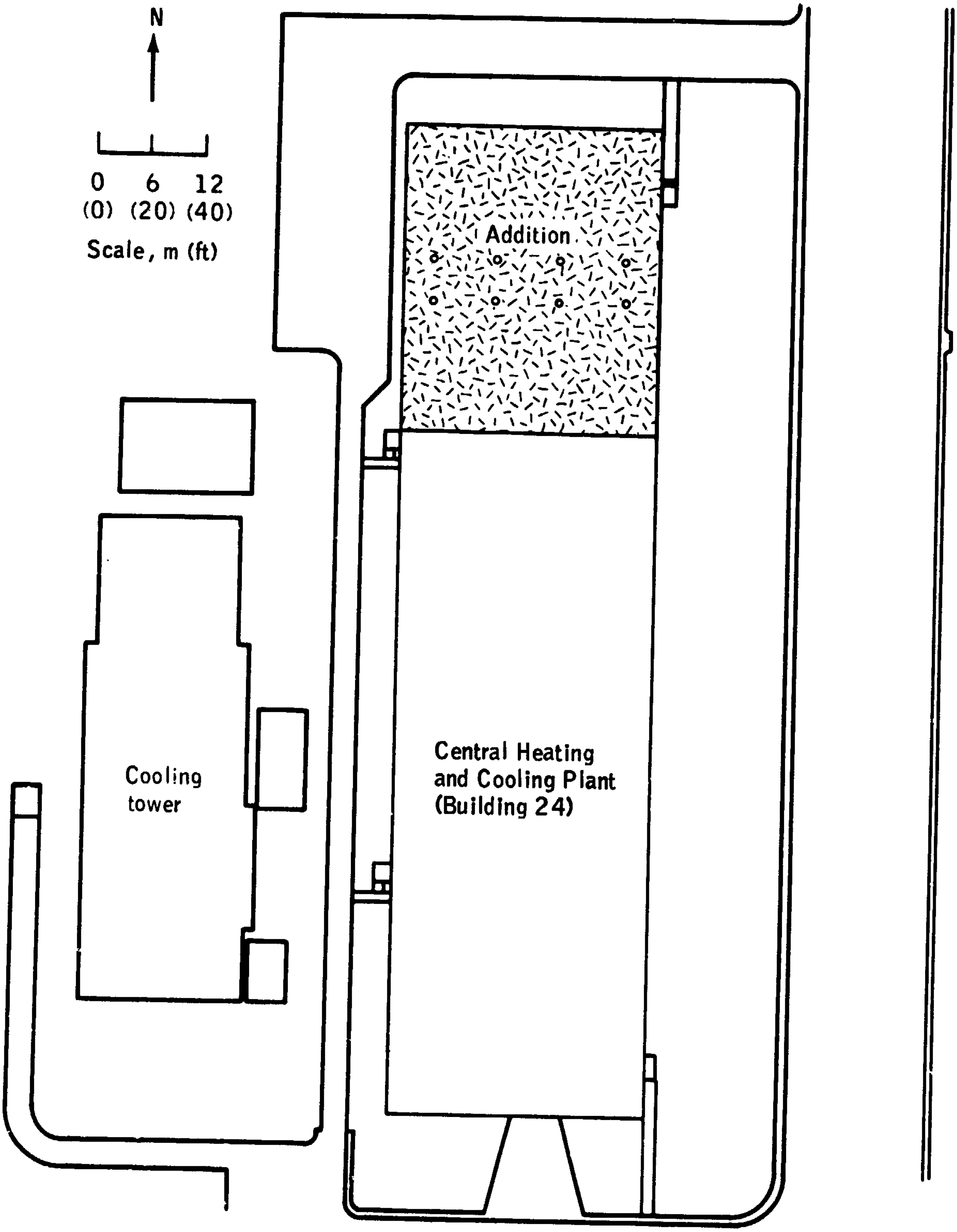


Figure 8.- Total energy system site plan.

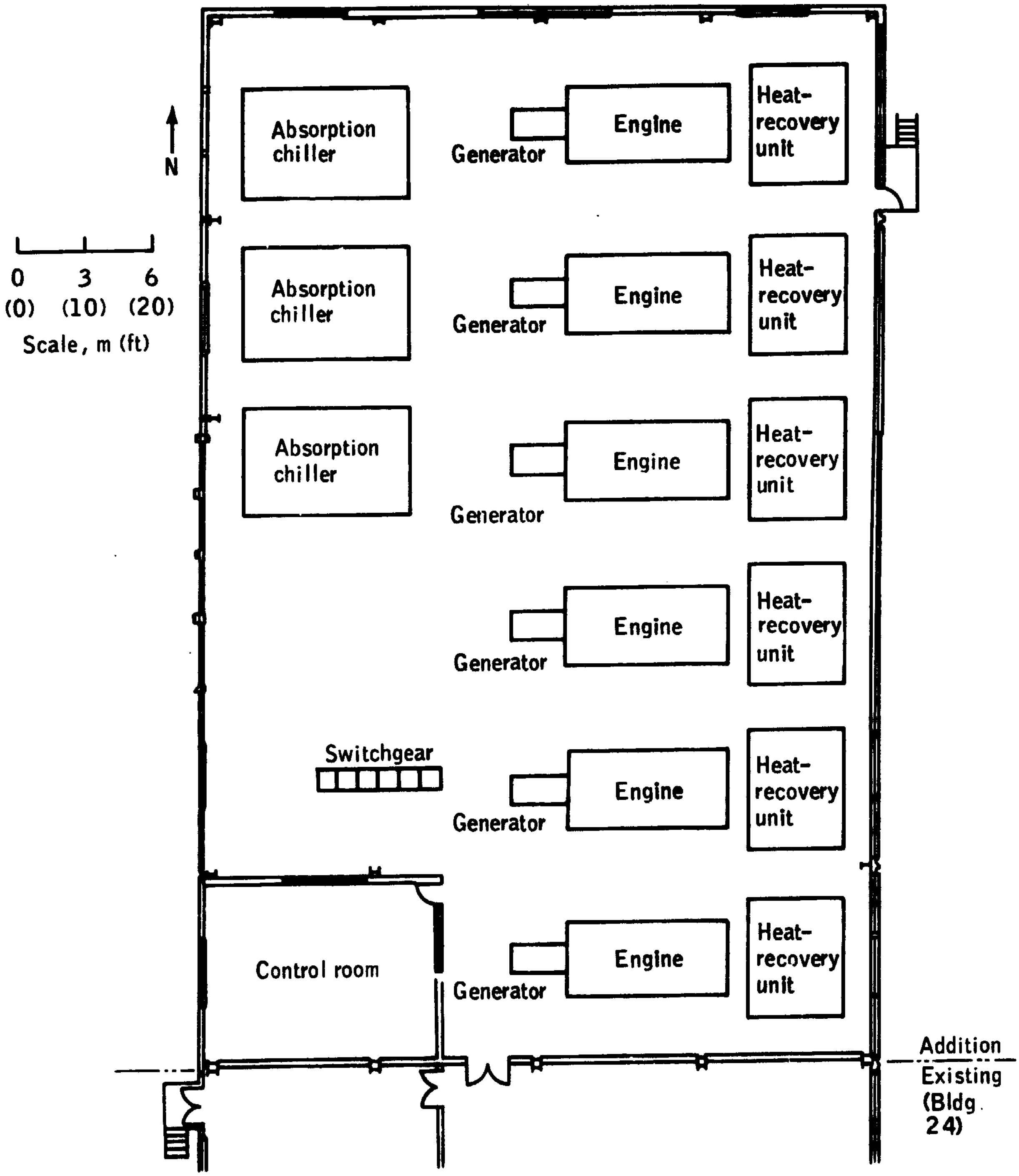


Figure 9.- Total energy system floor plan.

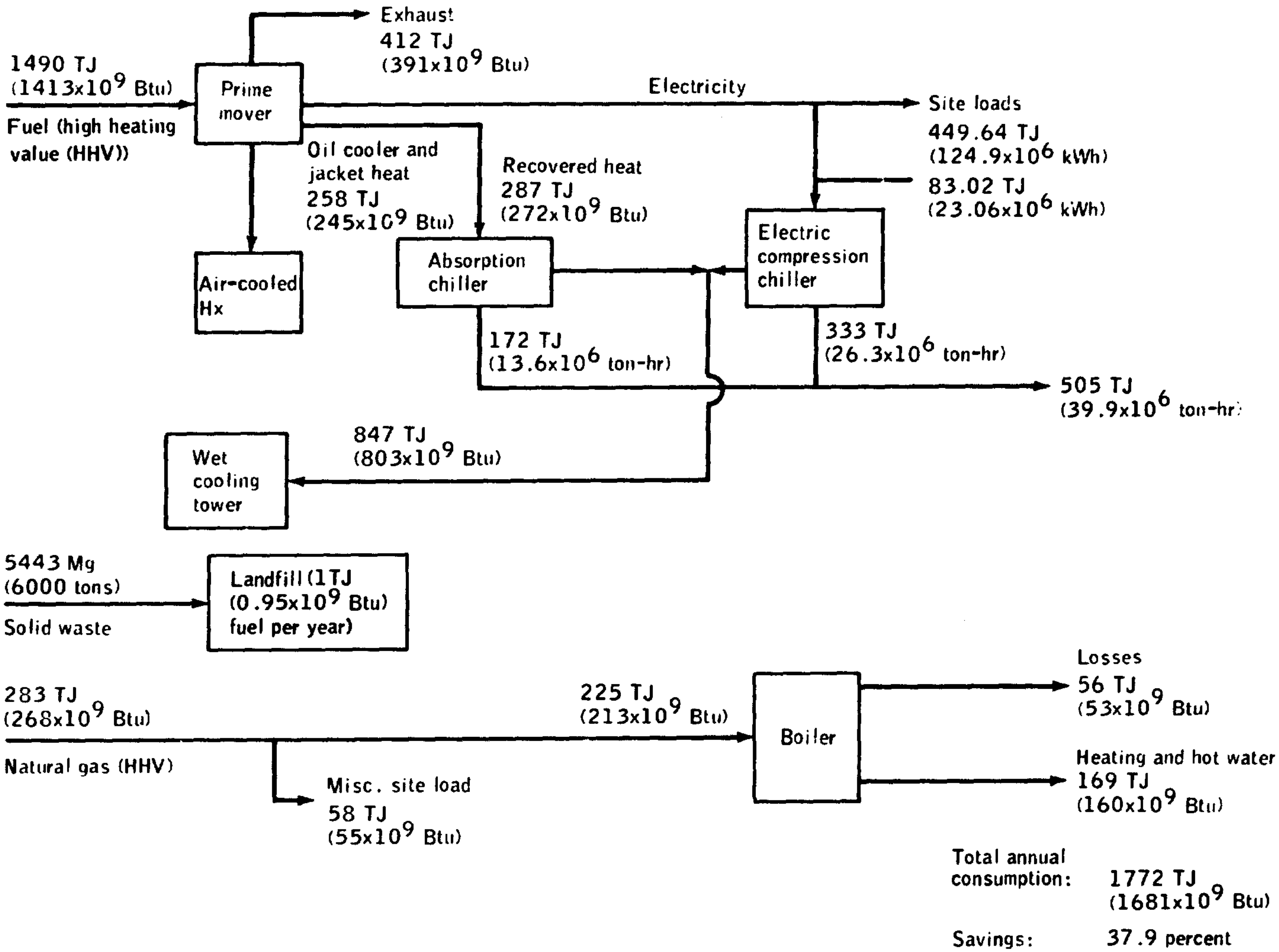


Figure 10.- Total energy system energy flow diagram, configuration 1: steam distribution; landfill solid waste.

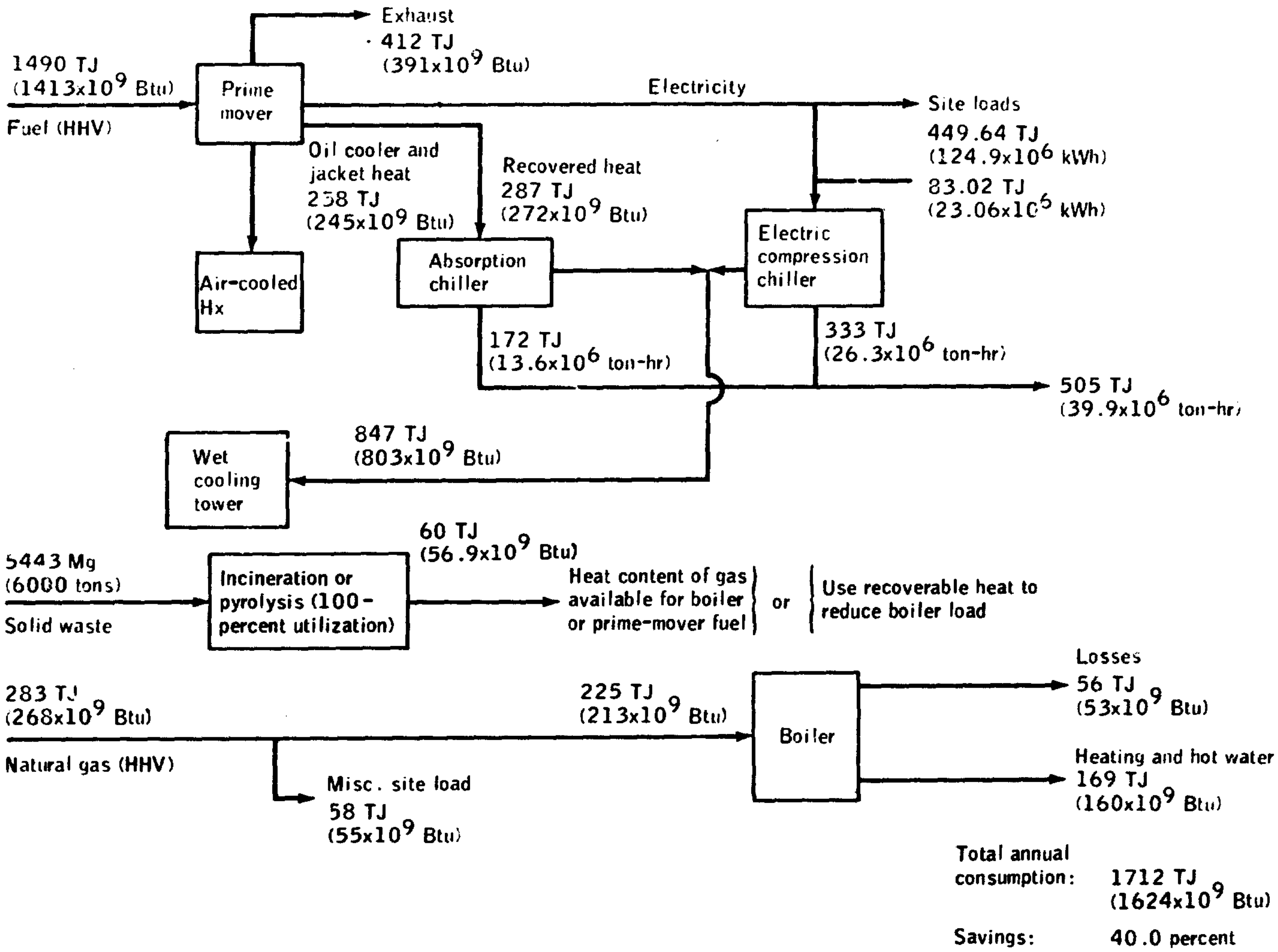


Figure 11.- Total energy system energy flow diagram, configurations 2 and 3: steam distribution; solid-waste incineration or pyrolysis.

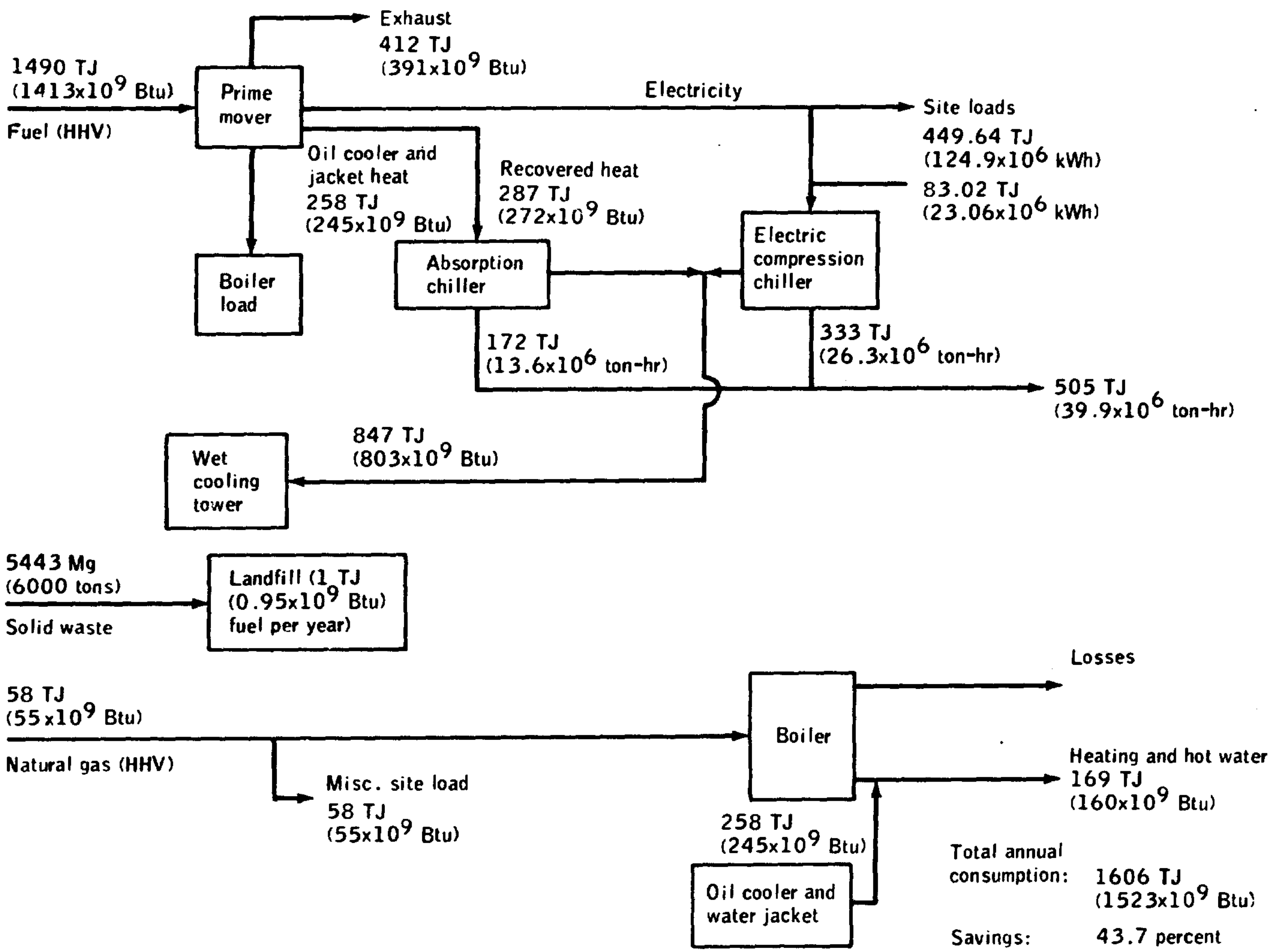


Figure 12.- Total energy system energy flow diagram, configuration 4: 353 K (175° F) hot-water distribution; landfill solid waste.

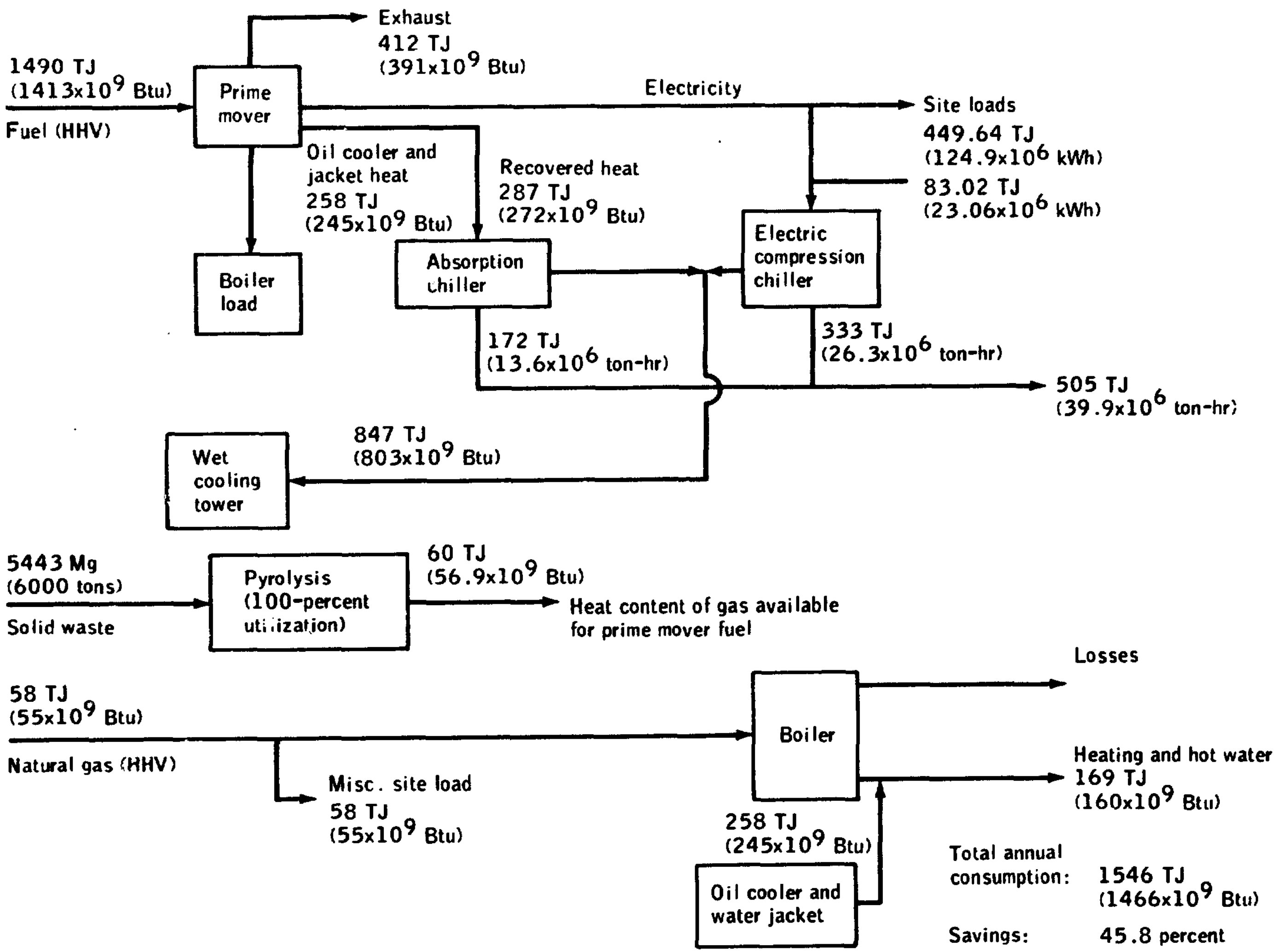


Figure 13.- Total energy system energy flow diagram, configuration 5: 353 K (175° F) hot-water distribution; pyrolysis of solid waste.

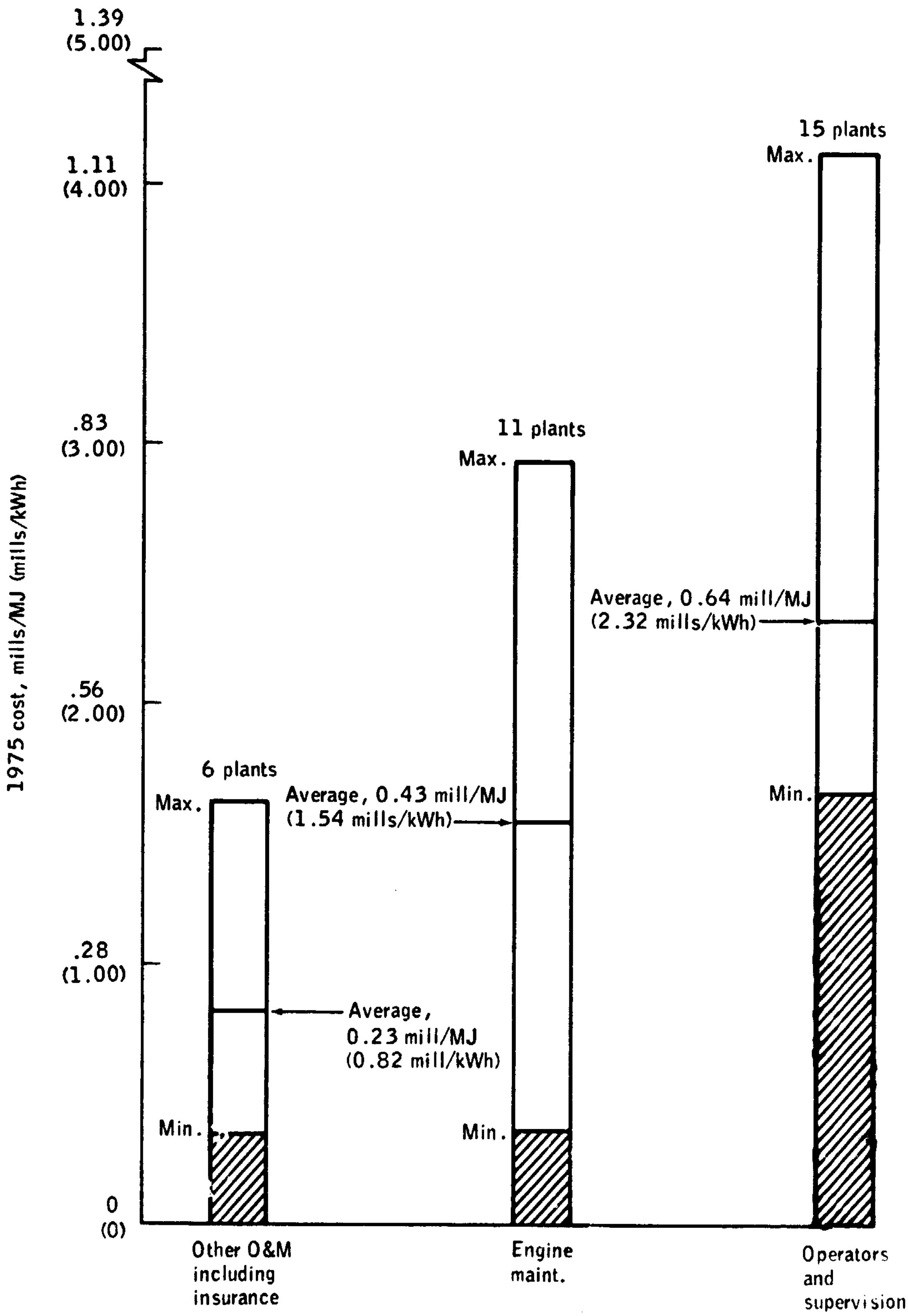


Figure 14.- Total energy plant O&M costs (based on the ASME 1972 report on diesel and gas engine power costs).

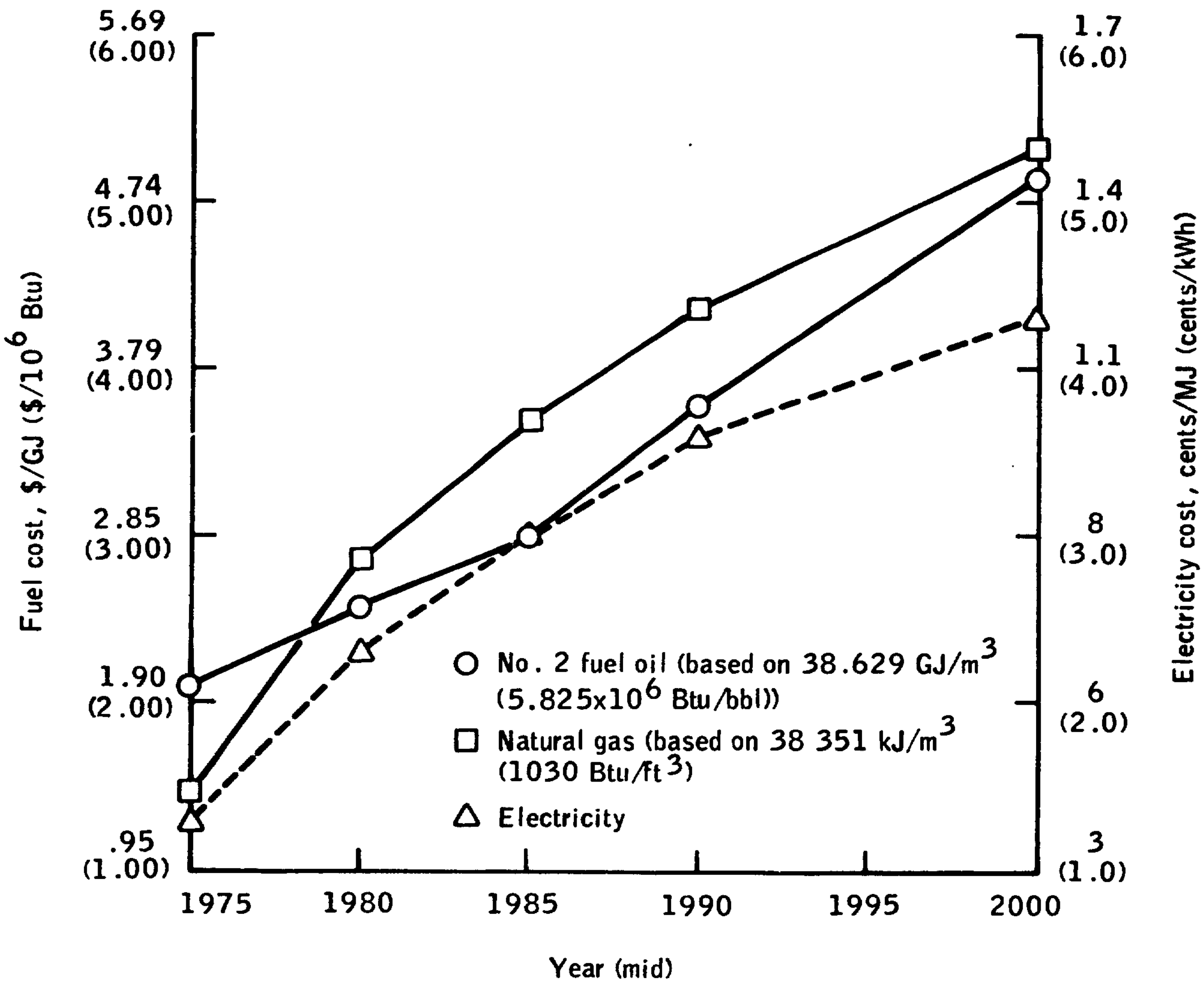


Figure 15.- Projected energy prices (baseline-case scenario by the Ralph M. Parsons Company, November 1975).

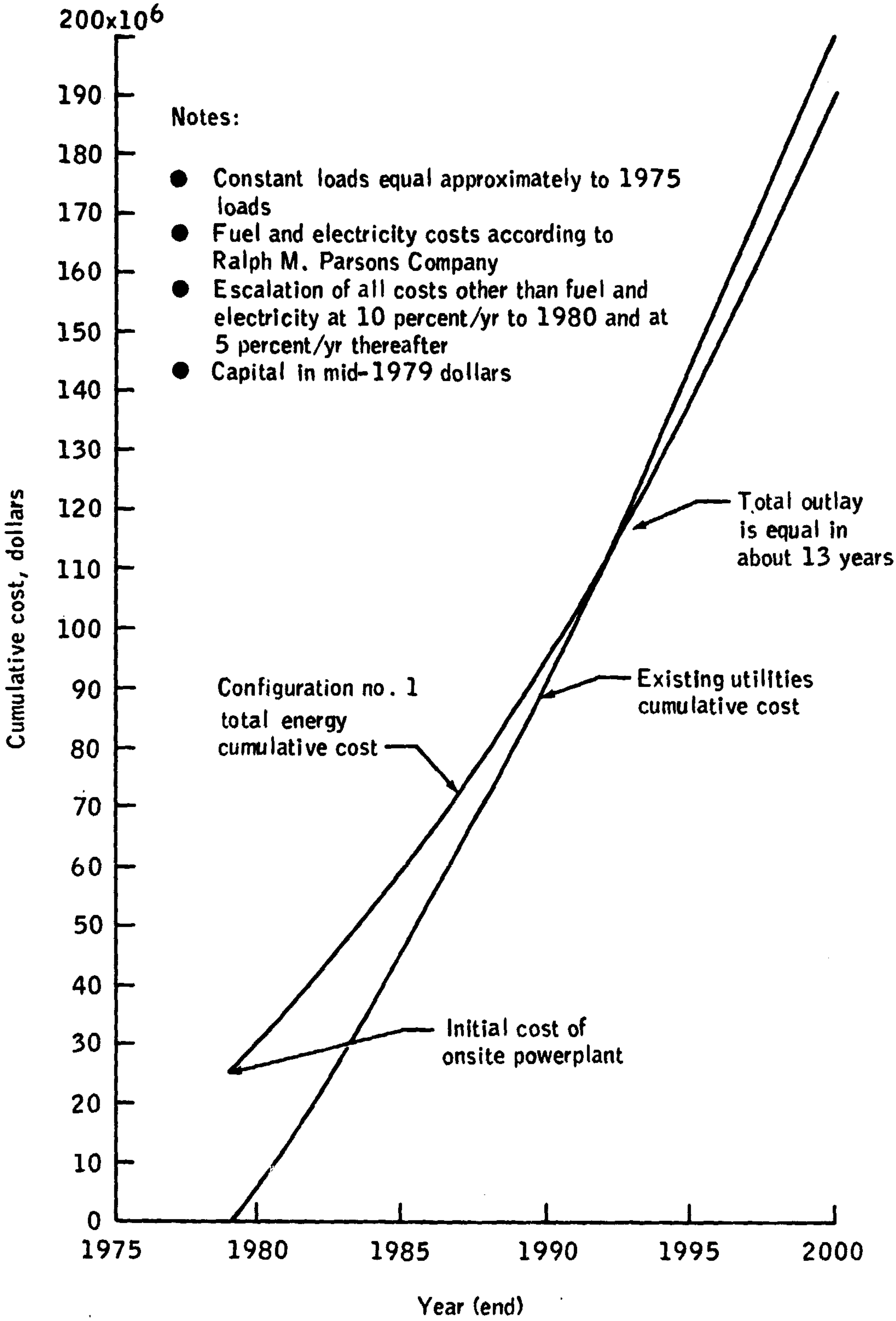


Figure 16.- Total energy concept (configuration 1) cost comparisons.

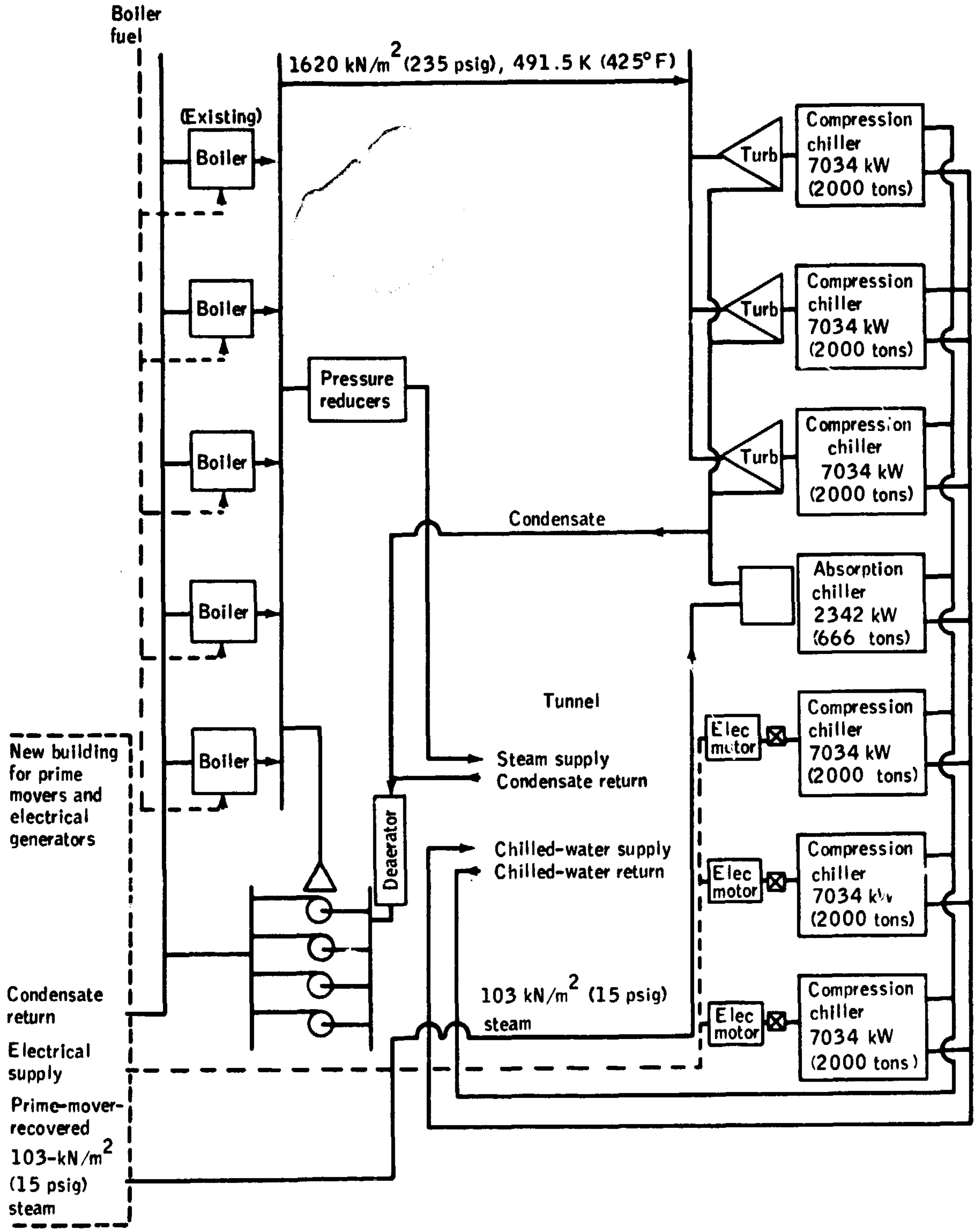


Figure 17.- Engine-generator/electric-motor-driven chiller schematic.

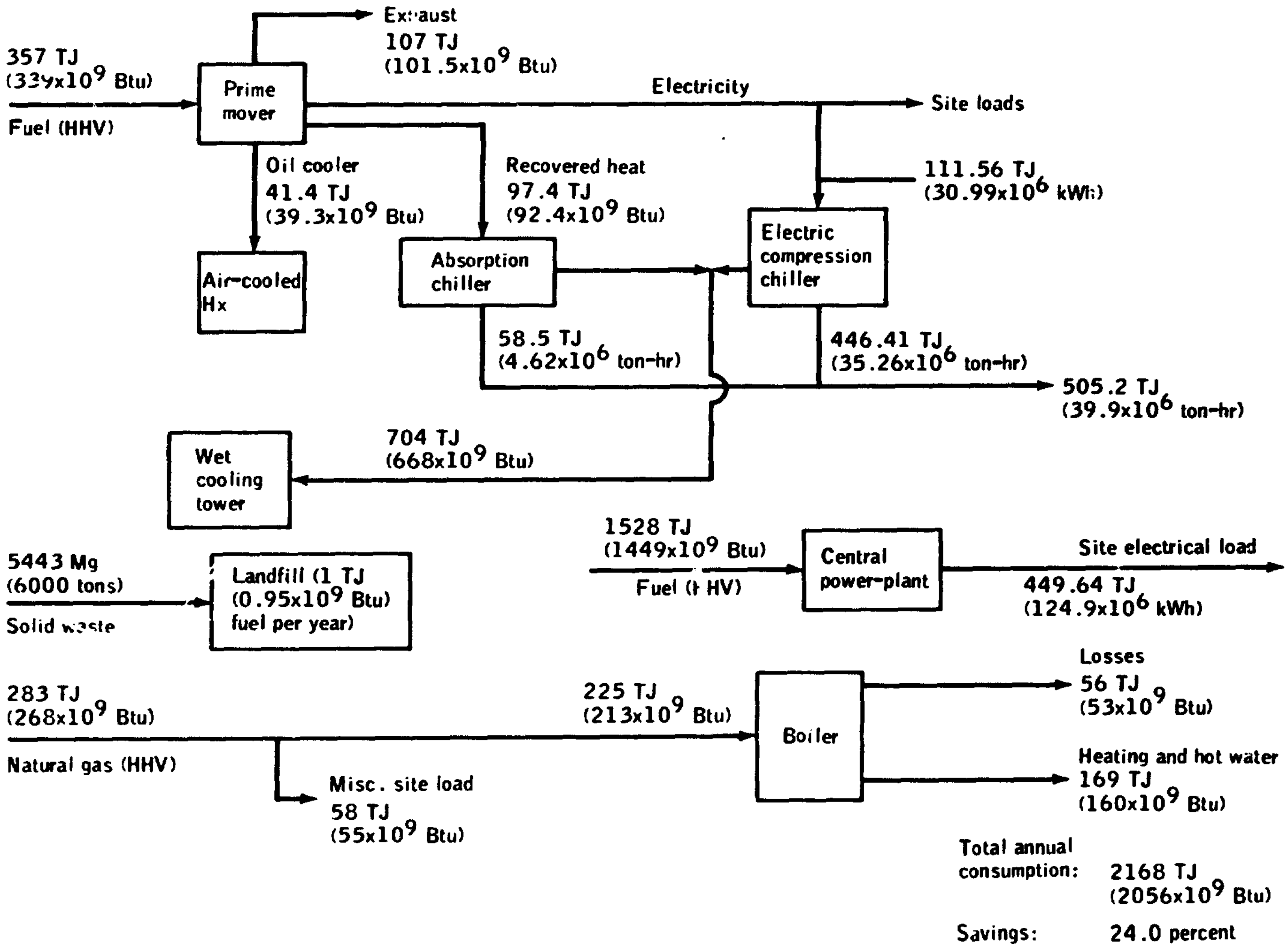


Figure 18.- Engine drive chiller option energy flow diagram, configuration 1: three York electric chillers for baseload; steam turbine chillers as required; landfill solid waste.

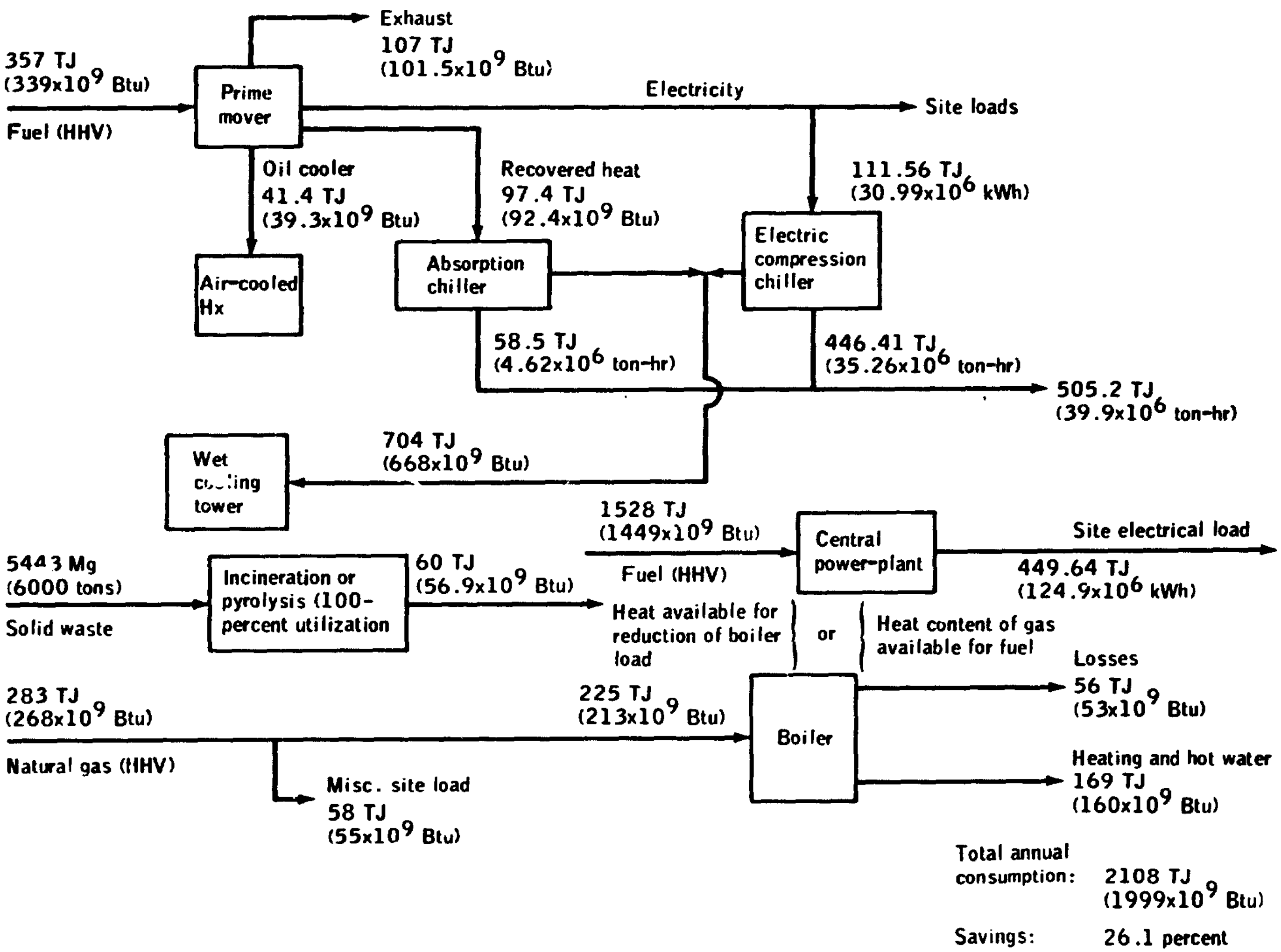
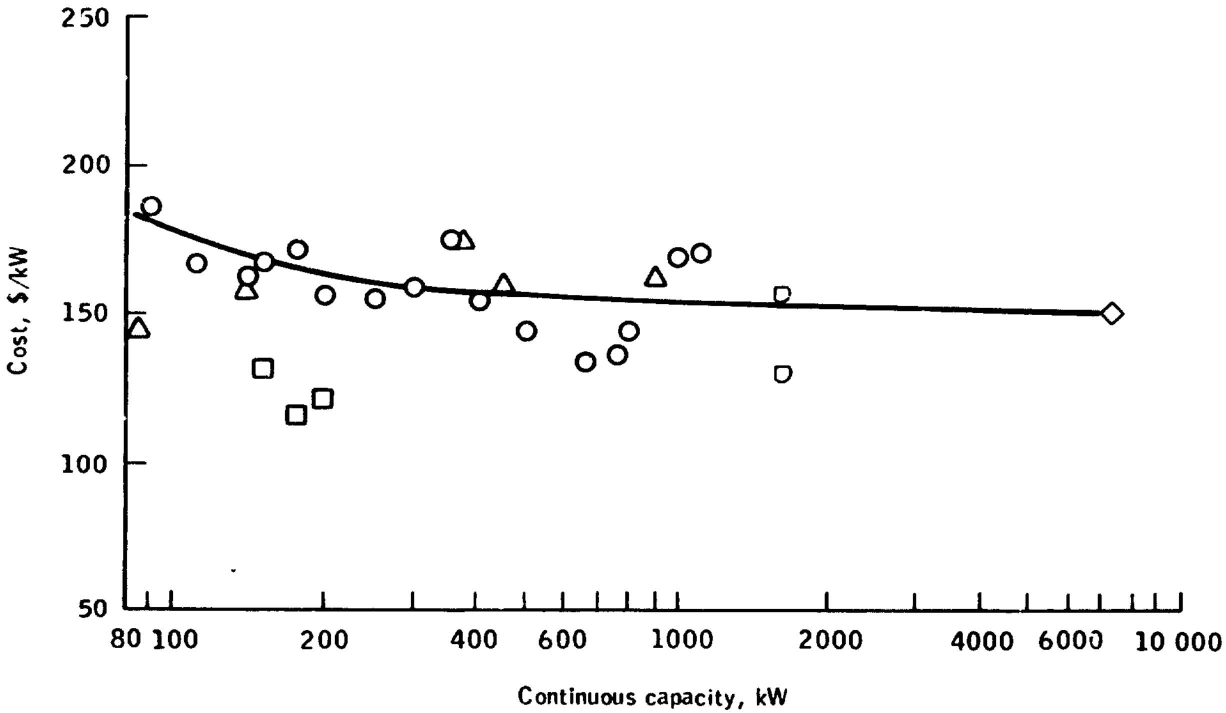


Figure 19.- Engine drive chiller option energy flow diagram, configurations 2 and 3: three York electric chillers for baseload; steam turbine chillers as required; solid-waste pyrolysis or incineration.



○ Diesel engine-generator sets with automatic transfer switch including fuel tank and radiator, 11 310 rad/min (1800 rpm), installed (1976 Building Cost File).

△ Gas or gasoline engine-generator sets including fuel tank and exhaust connection, installed (Means Building Construction Cost Data, 1975).

□ Allis-Chalmers 11 310-rad/min (1800 rpm) diesel engine-generator sets, no installation (1975 Richardson Process Plant Construction Estimating Standards).

◇ Estimated installed cost for JSC total energy concept.

D Estimated costs for engine-generator sets for JSC electric-motor-driven chiller concept: bottom point, engine-generators only; top point, engine-generators installed.

Figure 20.- Engine-generator unit costs, based on December 1975 prices.

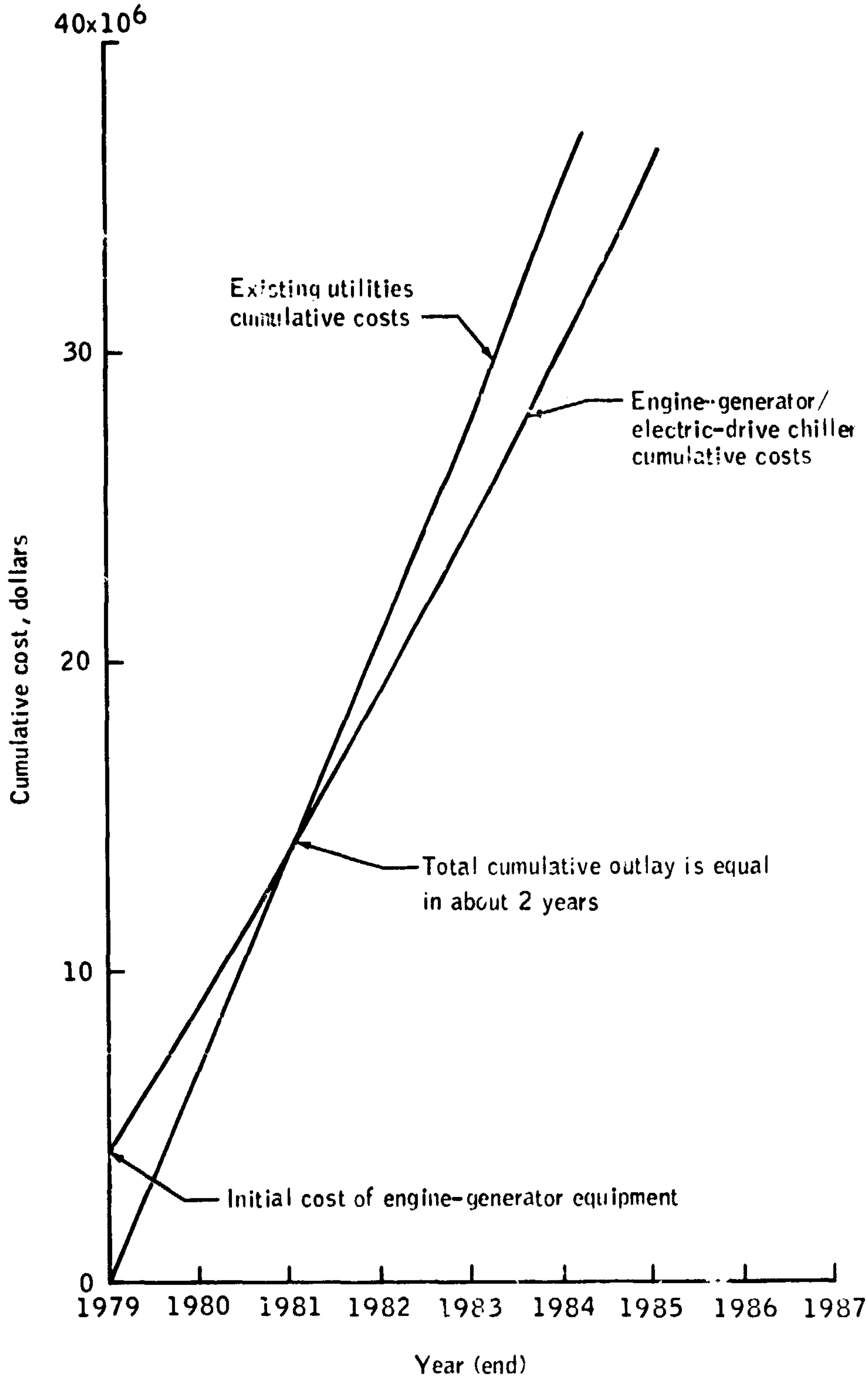


Figure 21.- Engine-generator/electric-drive chiller option (configuration 1) cost comparison.

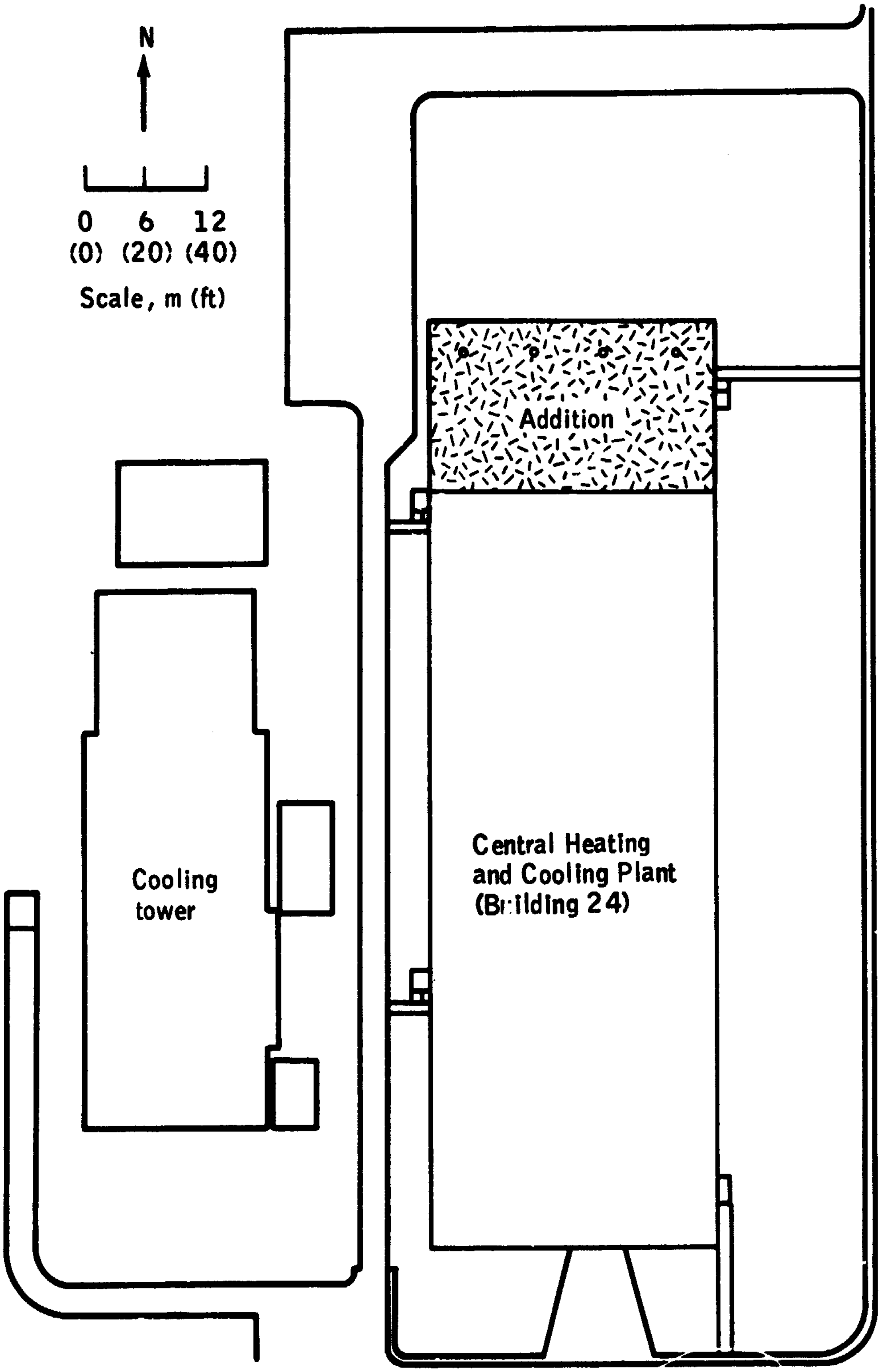


Figure 22.- Engine-driven chiller site plan.

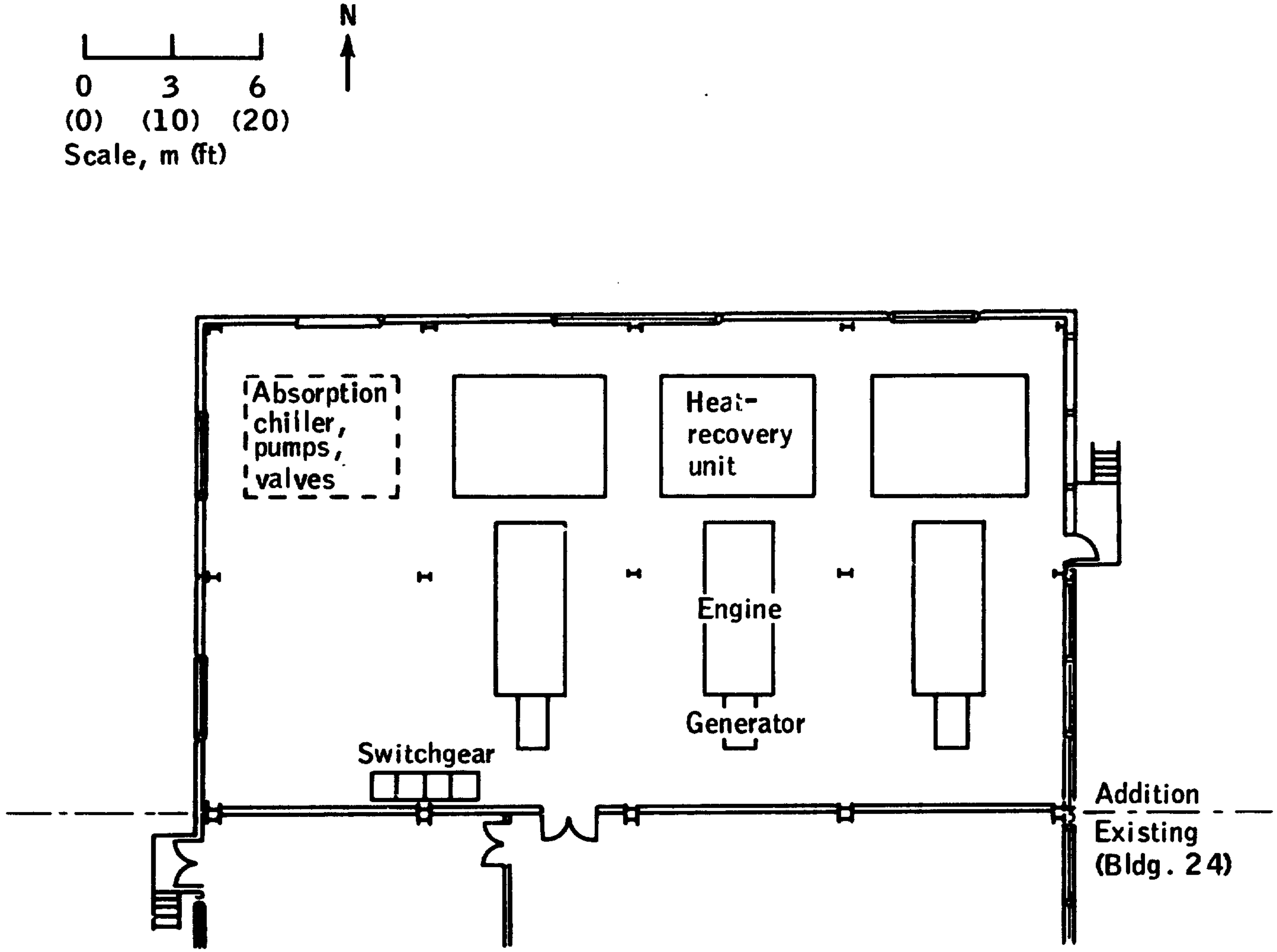


Figure 23.- Engine-driven chiller floor plan.

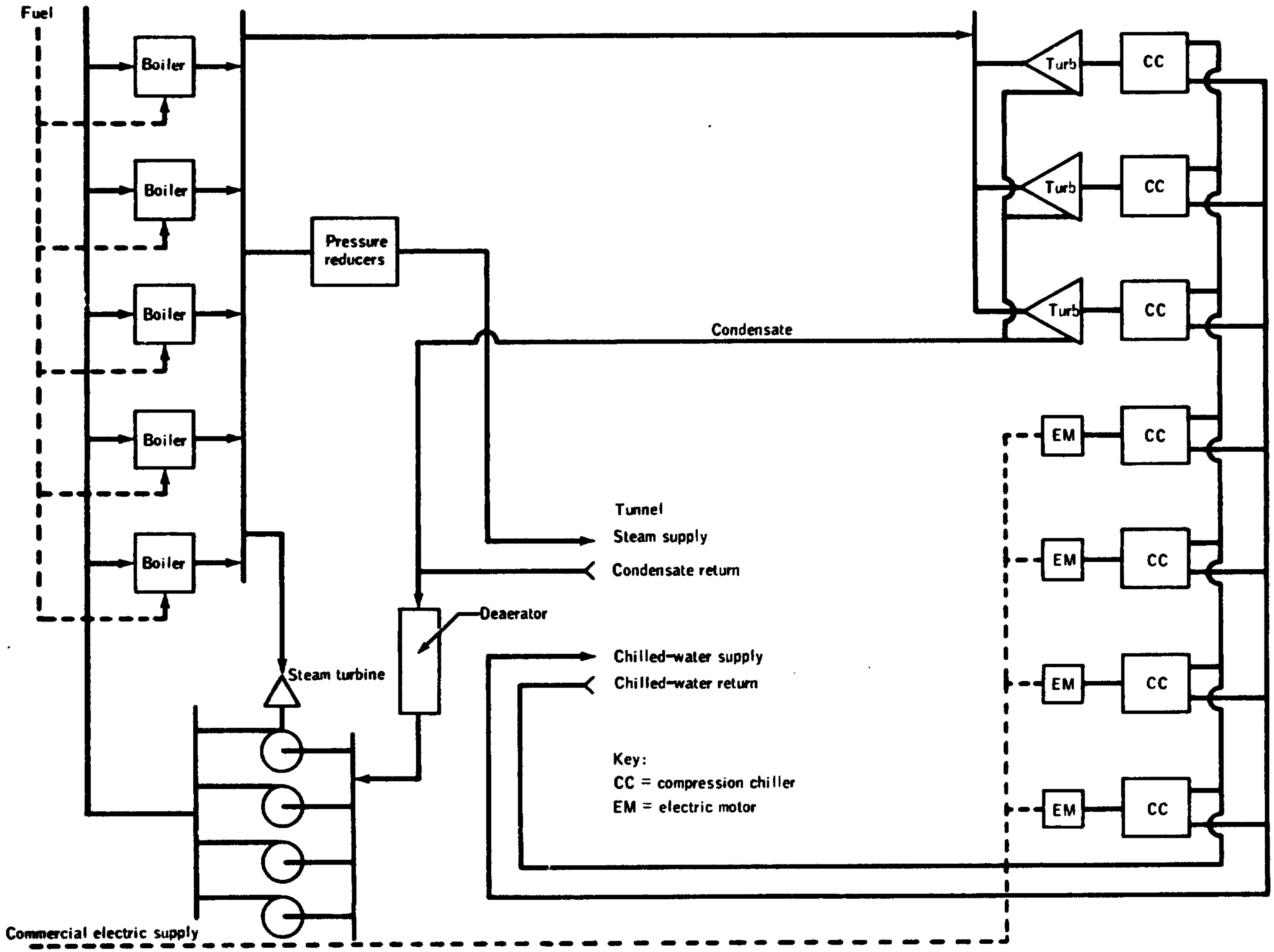


Figure 24.- Electric-drive chiller schematic.

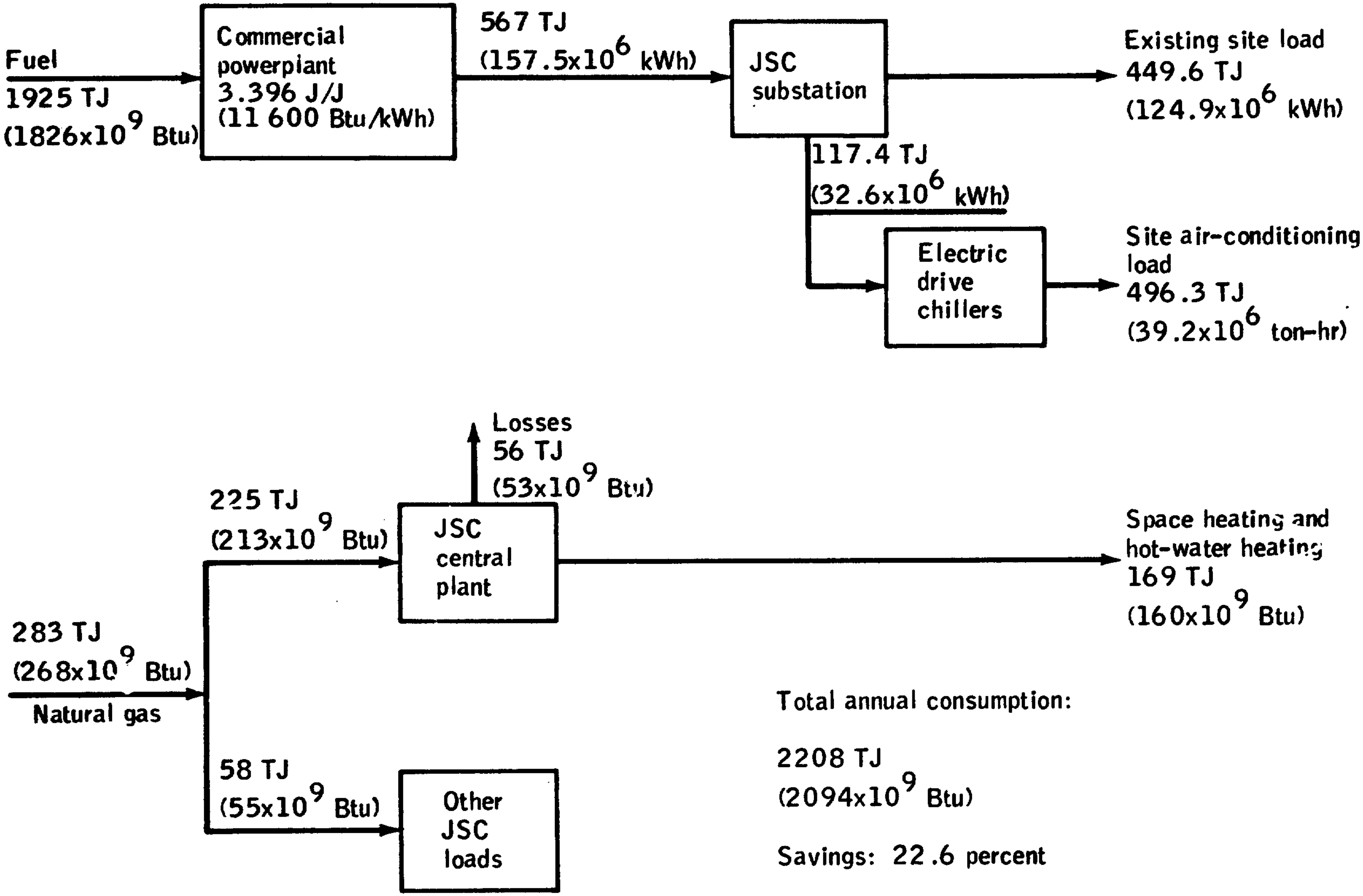


Figure 25.- Electric-drive chiller option energy flow diagram.

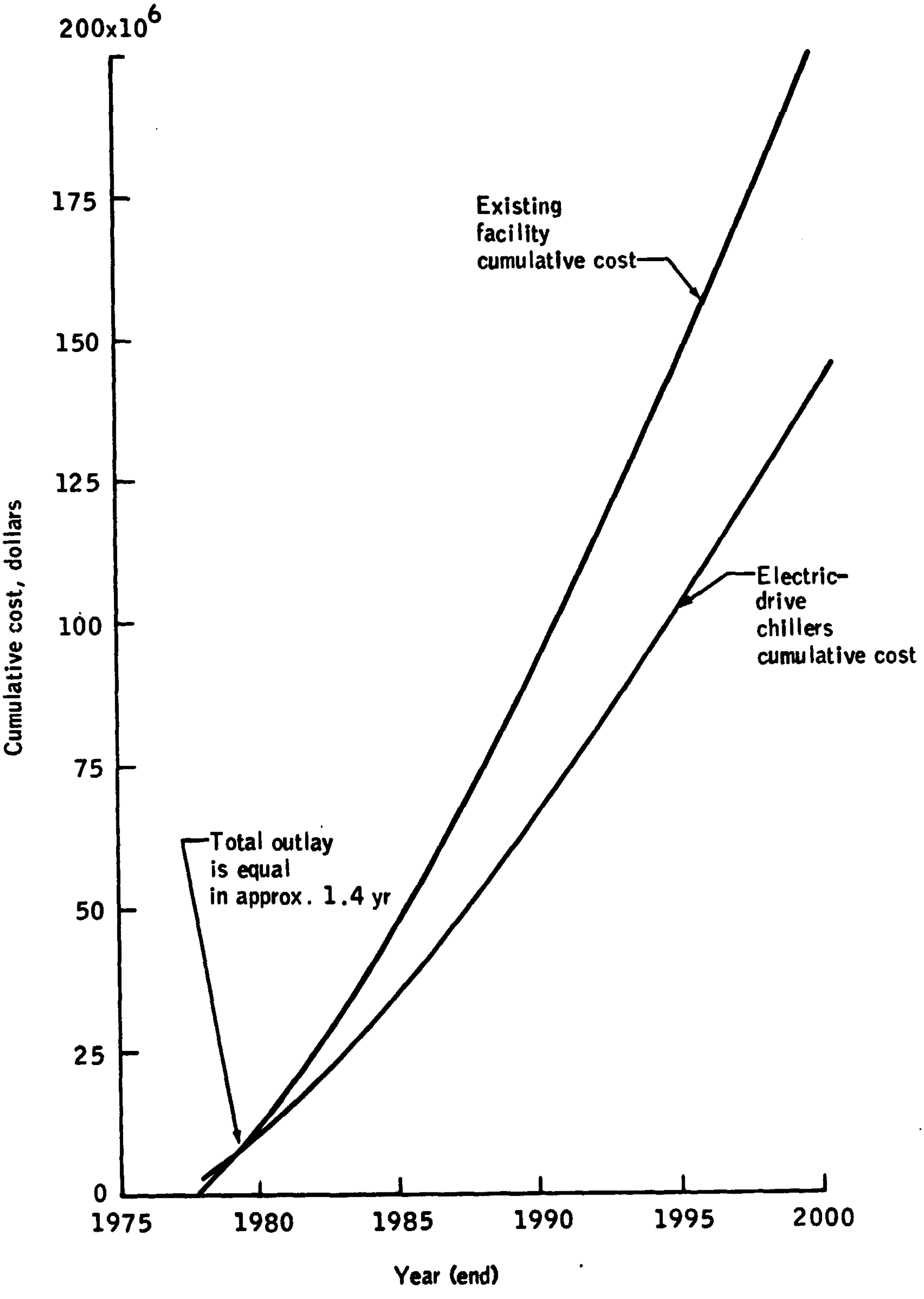


Figure 26.- Cumulative energy cost, electric-drive chiller compared to existing facility.

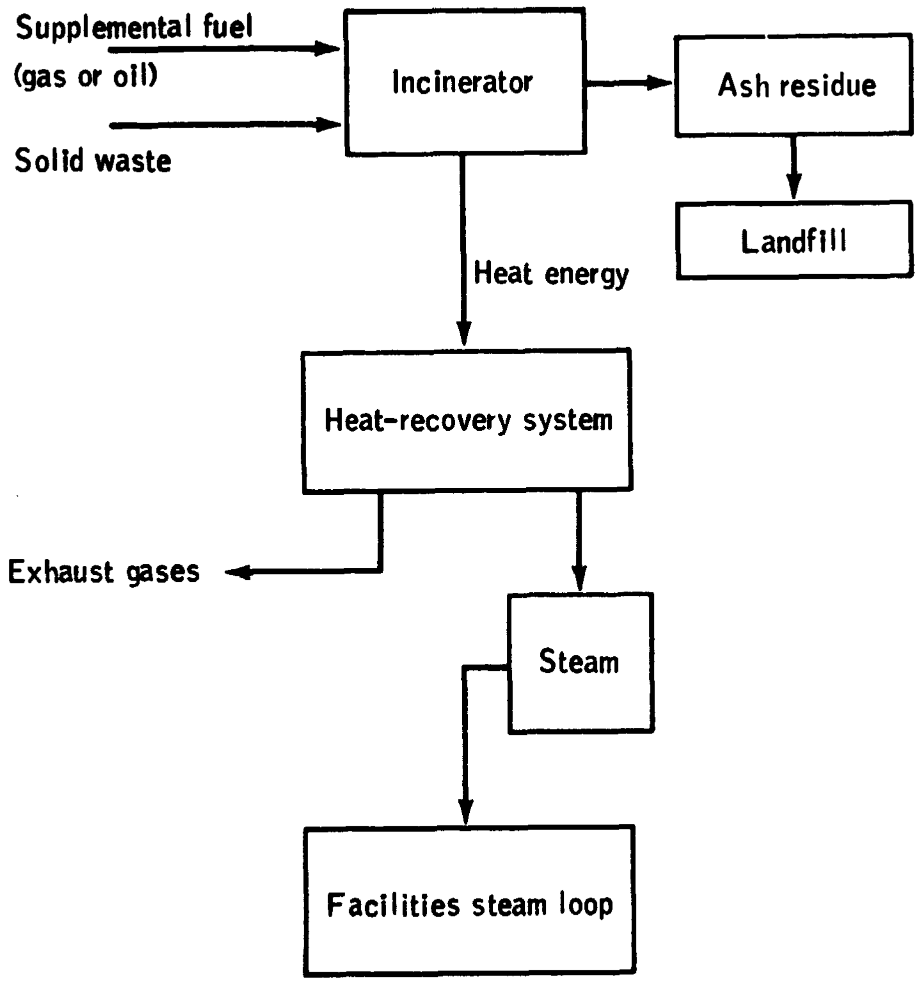


Figure 27.- Incinerator system flow.

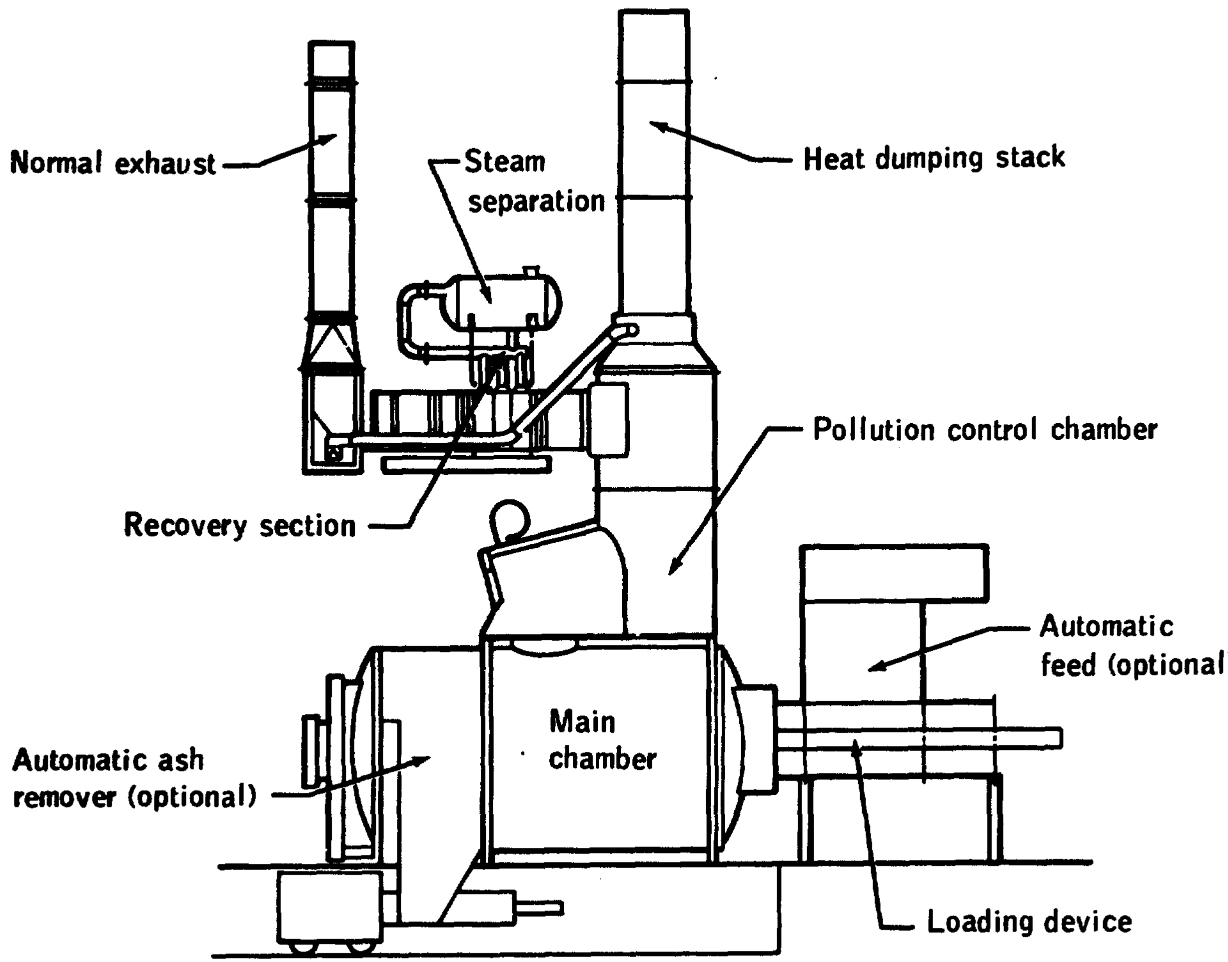


Figure 28.- Consumat incinerator.

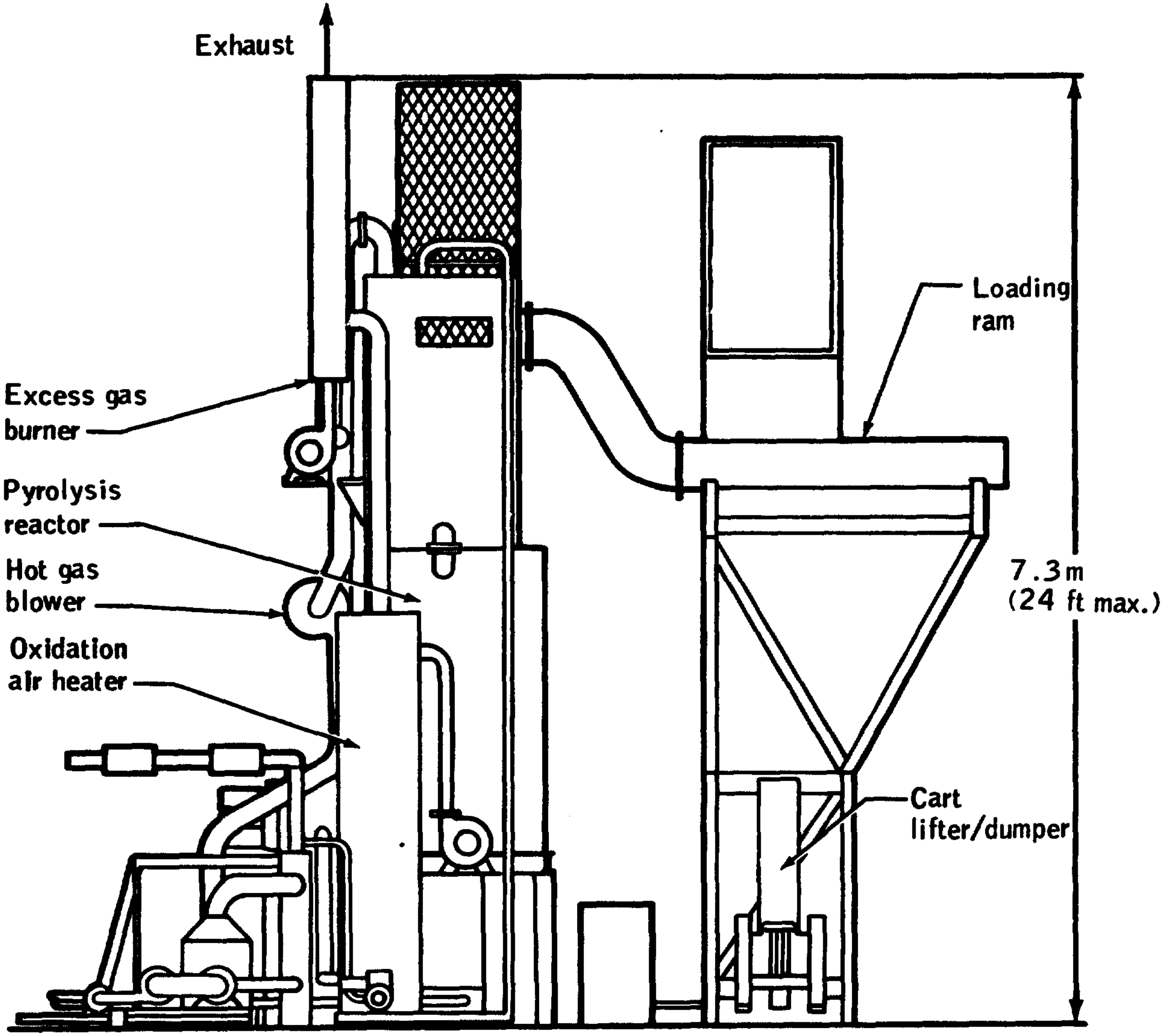


Figure 29.- The URDC pyrolysis system.

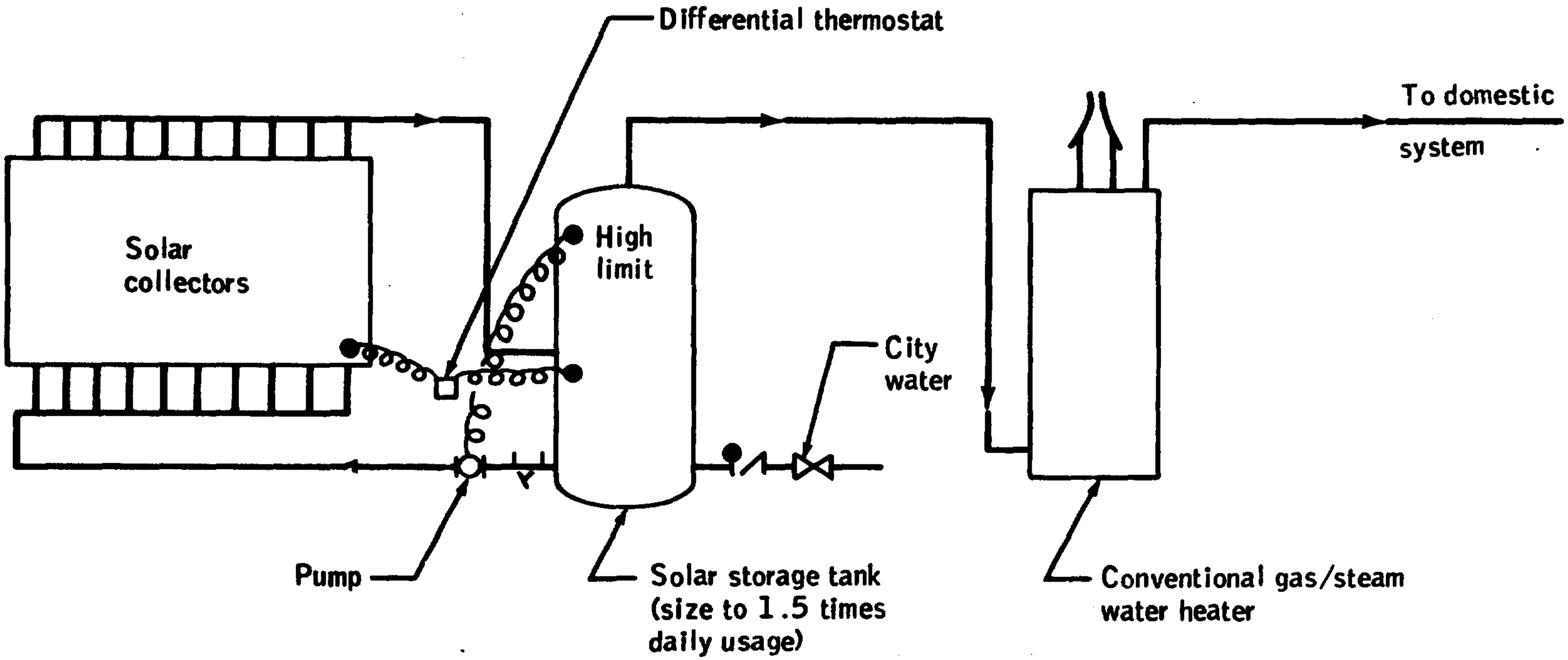


Figure 30.- Typical direct-solar-assist domestic-hot-water systcm.

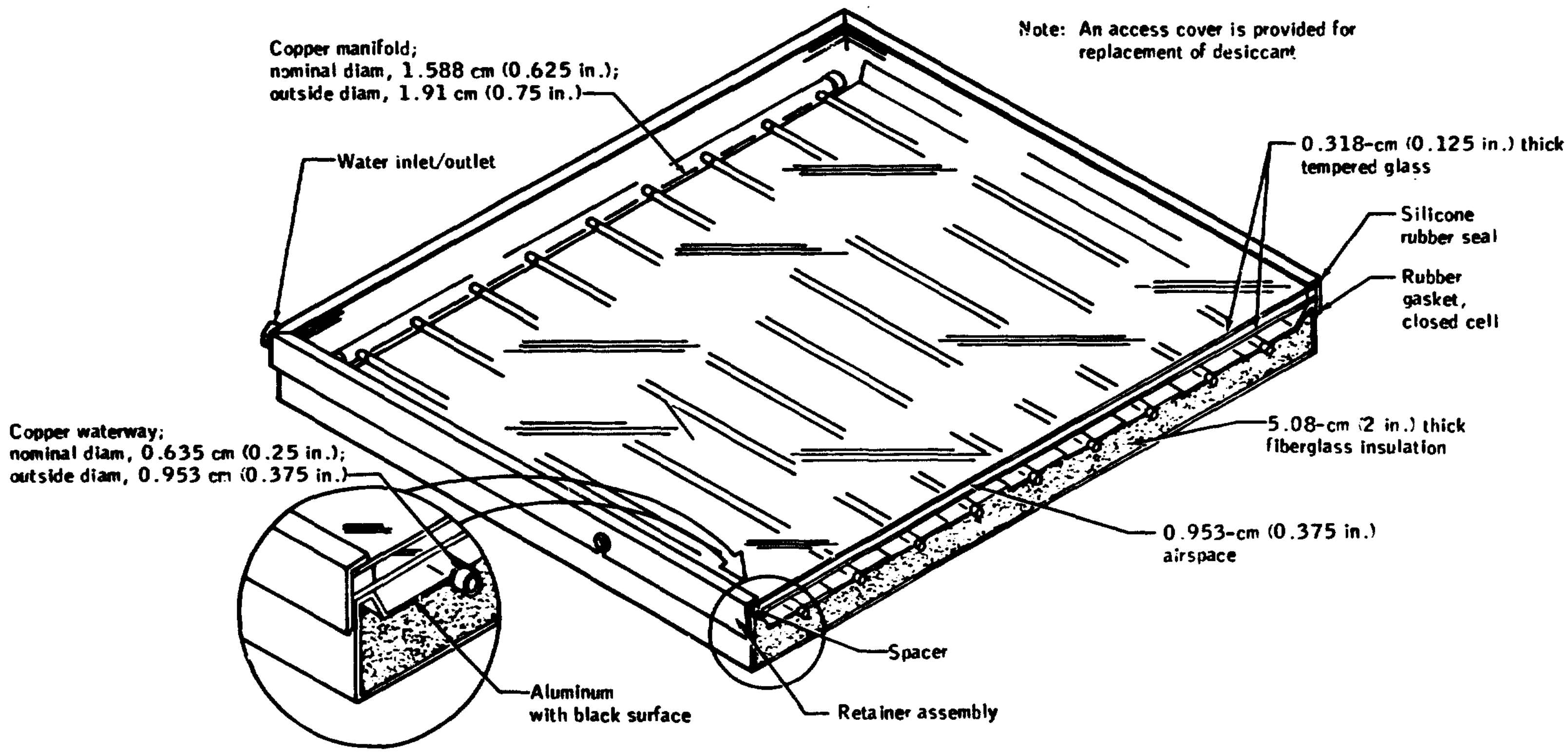


Figure 31.- Example of flat-plate solar collector (manufactured by Raypak).

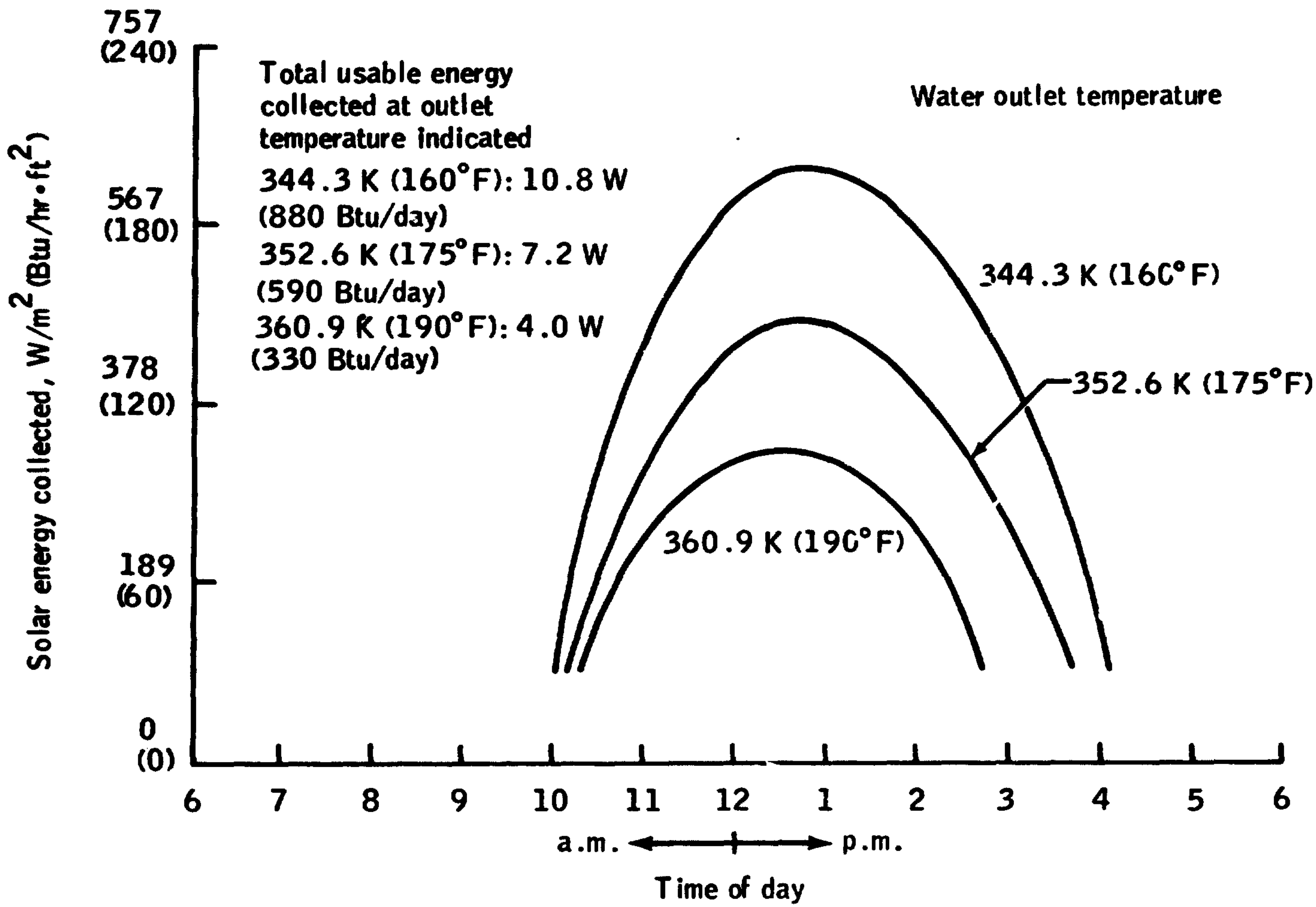


Figure 32.- Representative collectable solar energy based on water outlet temperature requirements (clear sky data).

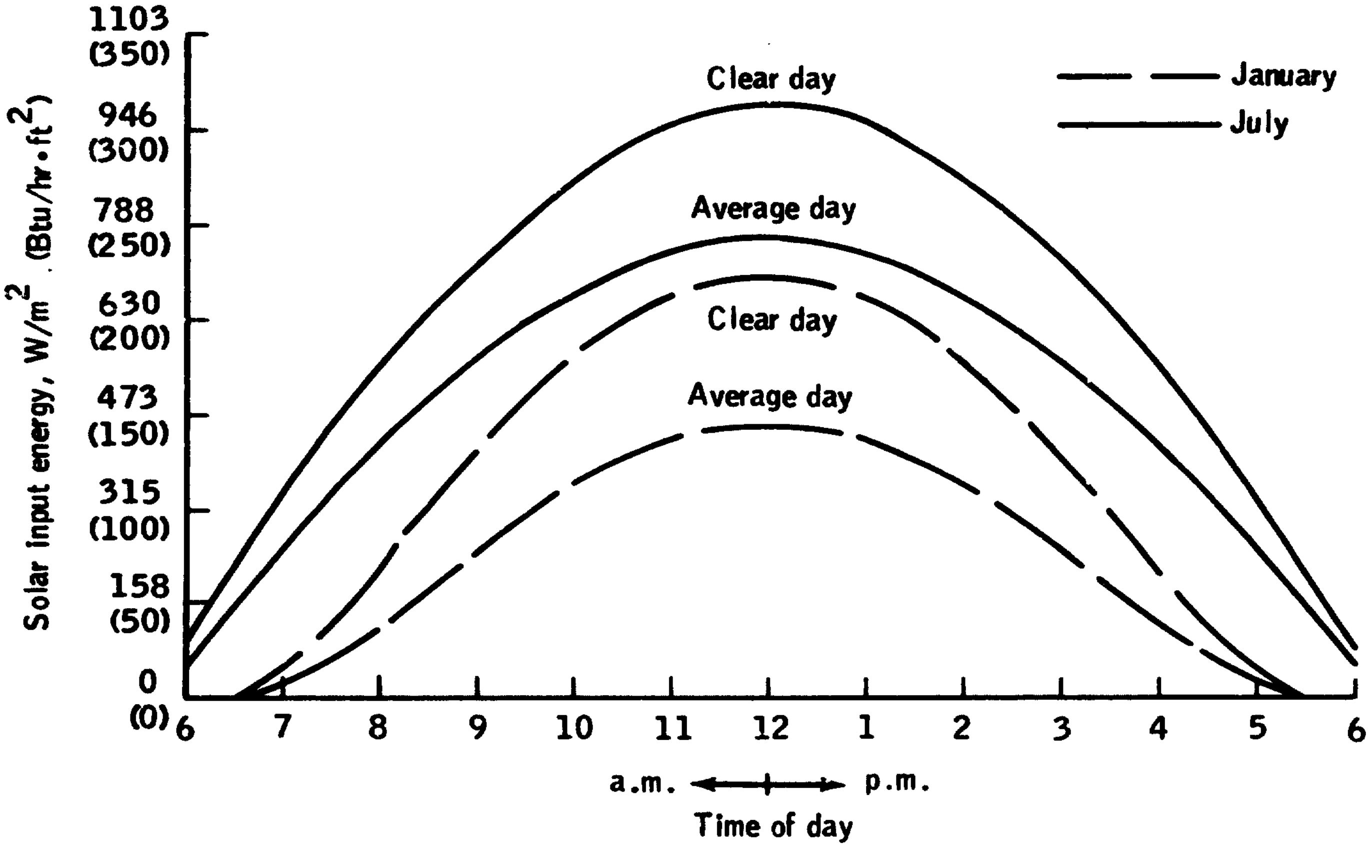


Figure 33.- Average day insolation for extreme months in the Houston area (latitude 30 N) on horizontal surfaces.

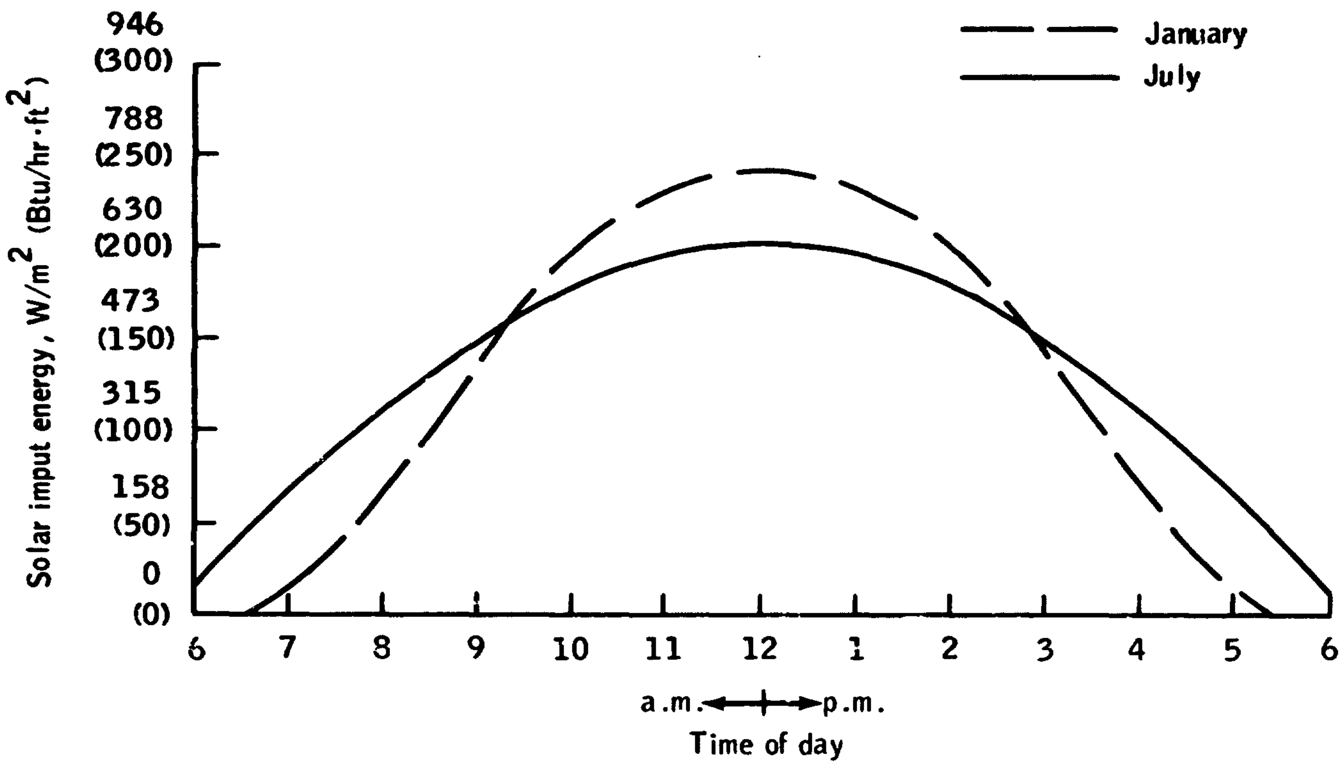


Figure 34.- Average day insolation for extreme months in the Houston area (latitude 30° N) with collectors tilted 29.5° to the horizontal.

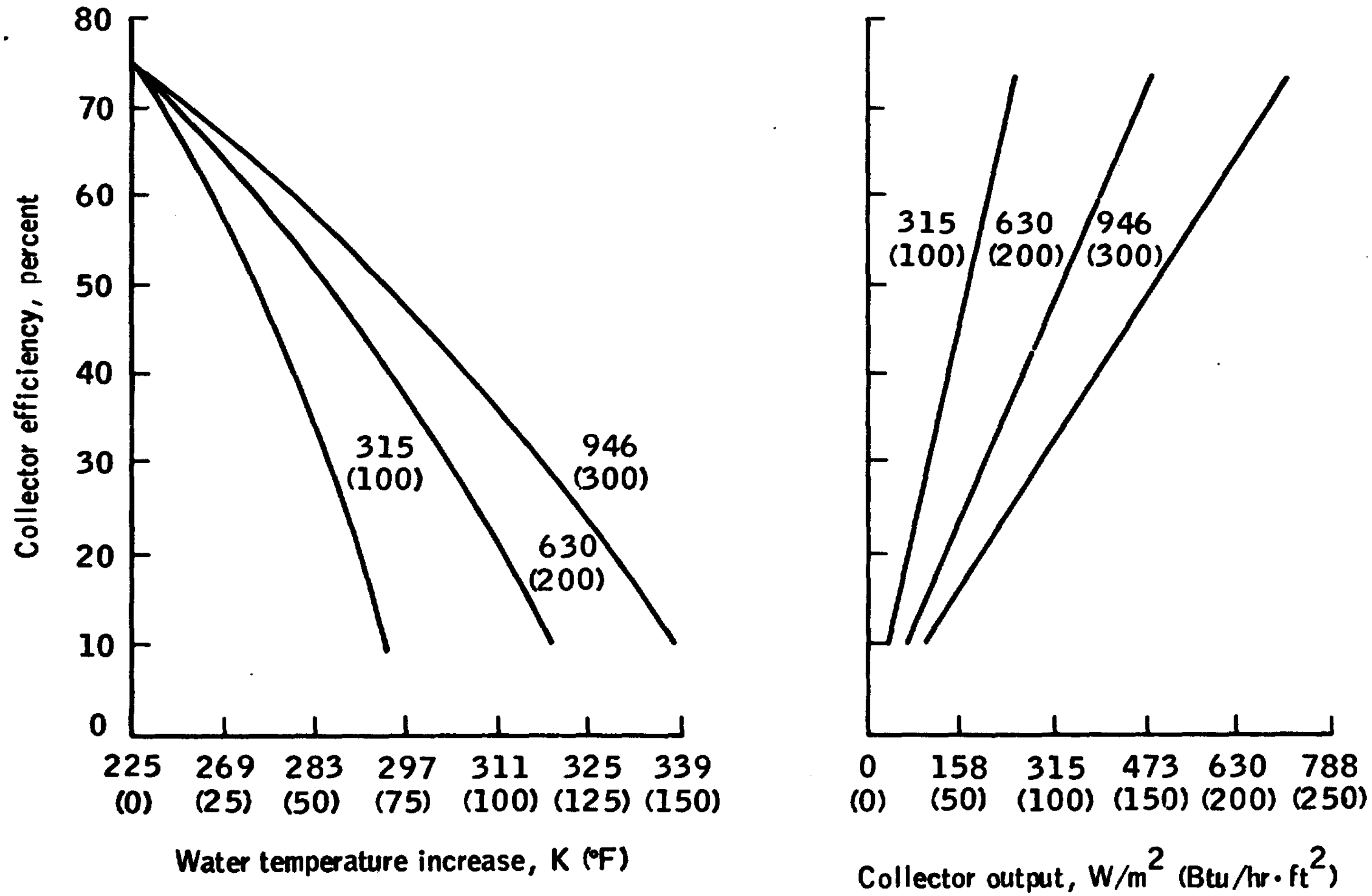


Figure 35.- Solar energy collector system rating (Revere collector with single glass glazing). Curve labels are solar input in watts per square meter (British thermal units per hour square foot).

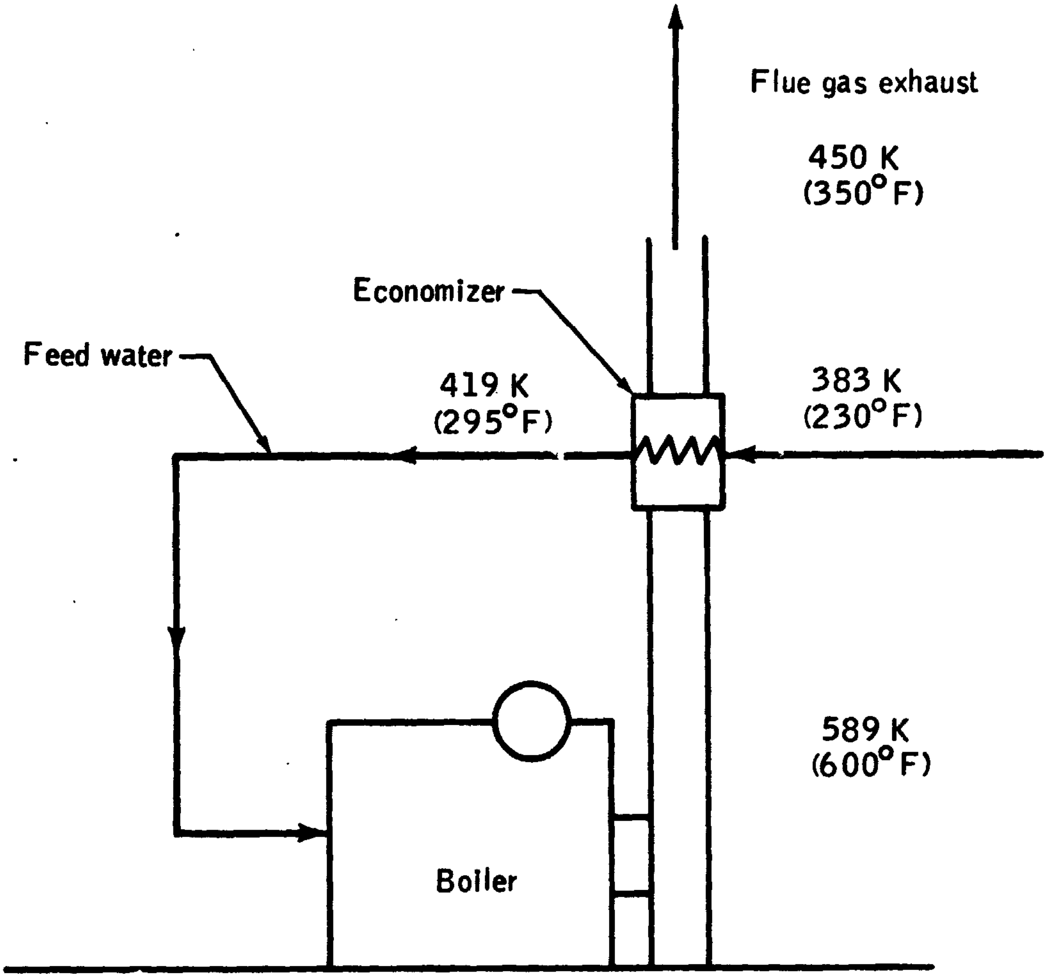


Figure 36.- Typical boiler economizer installation.

APPENDIX A

CENTRAL HEATING AND COOLING PLANT (BUILDING 24)

MODEL AND COMPUTER PROGRAM

As a part of this study, a computer program was developed to analyze energy flow and equipment performance in the existing building 24 central heating and cooling plant. The program also contains optional subroutines for analyzing electric-drive chillers in place of the existing steam-turbine-driven chillers. The program has been documented and is described in a contractor report.[7]

Shown in figure A-1 is a computer program mathematical model that illustrates the thermodynamic state-point conditions used. Table A-I is a sample calculation resulting if a building 24 modification were made to replace the steam-turbine-driven chillers with two electric-drive chillers of 7034 kilowatts (2000 tons) each. In this instance, the total energy saving would be 22.75 percent and savings on boiler fuel would be 34.03 percent.

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[7]Computer Program for Energy Analysis of NASA-JSC Central Heating and Cooling Plant (Building 24). Lockheed Electronics Company, Inc., TM-5054, October 1975.

TABLE A-I.- FUEL, ENERGY, AND COST SAVINGS WITH ELECTRIC CHILLERS

Time	Boiler fuel savings, percent	Energy savings, percent	Cost savings, percent
1 a.m.	43.68	31.26	23.05
2 a.m.	30.15	21.18	14.20
3 a.m.	30.15	21.20	14.23
4 a.m.	40.31	29.33	19.15
5 a.m.	30.52	20.66	12.98
6 a.m.	41.38	28.56	20.47
7 a.m.	31.55	20.70	11.82
8 a.m.	32.85	20.43	12.05
9 a.m.	35.16	22.23	13.40
10 a.m.	32.30	21.25	13.49
11 a.m.	32.40	21.24	13.45
12 m.	32.39	20.57	12.74
1 p.m.	32.37	20.41	12.60
2 p.m.	35.63	21.76	12.86
3.p.m.	31.90	20.91	13.24
4 p.m.	33.04	21.64	13.69
5 p.m.	32.88	22.24	14.39
6 p.m.	32.78	22.54	14.79
7 p.m.	33.30	22.14	13.77
8 p.m.	30.15	19.62	11.81
9 p.m.	30.59	20.10	12.24
10 p.m.	40.06	27.09	17.54
11 p.m.	30.15	19.80	12.11
12 p.m.	41.12	29.14	19.70
Hourly average	34.03	22.75	14.57

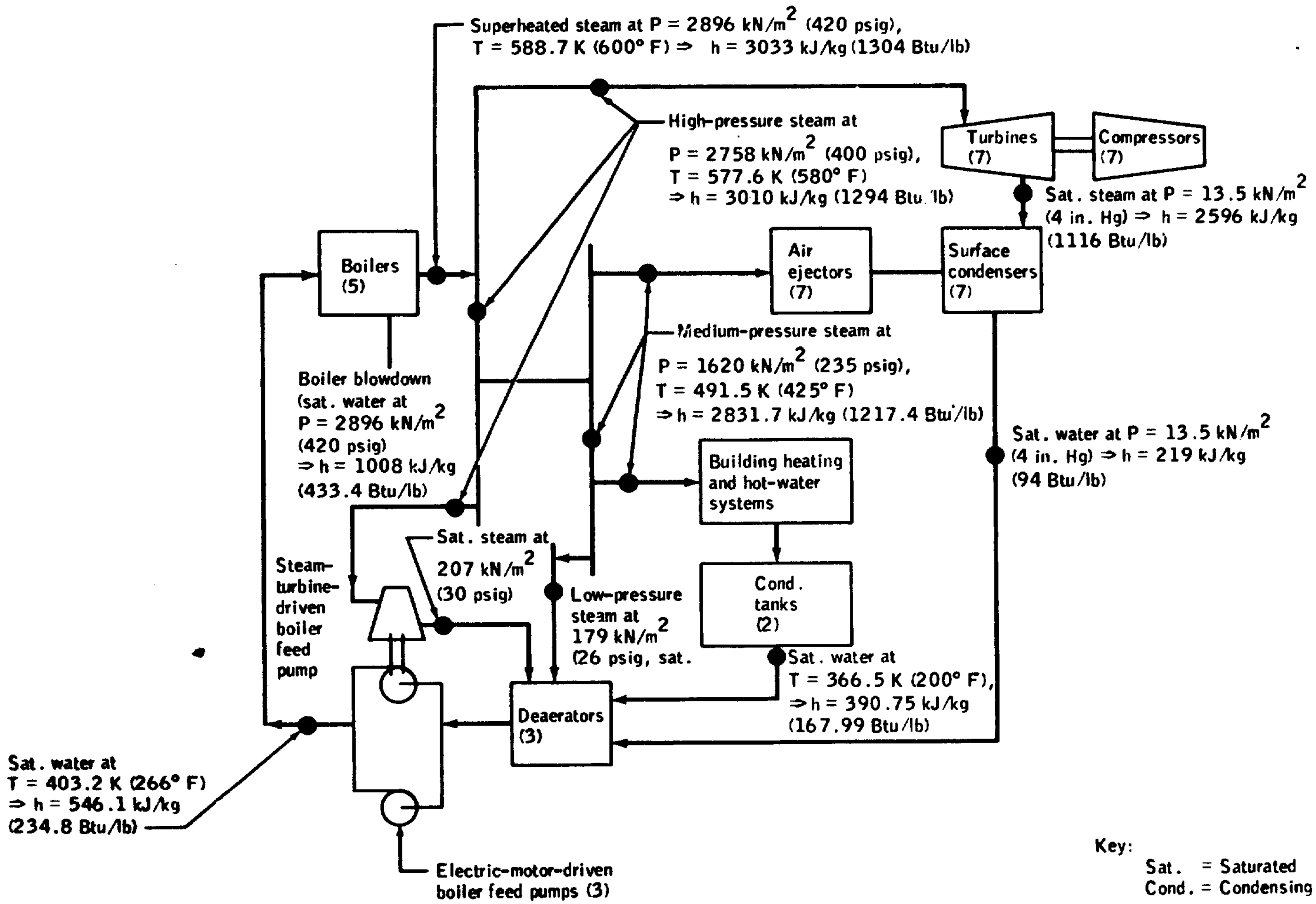

Figure A-1.- Enthalpy (state-point conditions) used in building 24 program, where P is pressure, T is temperature, and h is specific enthalpy.

APPENDIX B

BUILDING AND SITE HEATING AND COOLING ENERGY CONSERVATION

A study was performed in January 1976 to evaluate architectural design and building system modifications for heating- and cooling-load reduction and energy conservation. This study was performed by an architectural firm and an NASA contractor under the direction of the NASA Lyndon B. Johnson Space Center (JSC) Urban Systems Project Office. The methodology employed in the study was as follows.

1. Select a baseline facility model for JSC buildings to determine heating, air-conditioning, and electrical loads. The model was subdivided into standard zones for loads analysis.

2. Develop conceptual designs to reduce energy consumption for each zone.

3. Analyze the extent of potential application of conceptual designs to a JSC site. The site was analyzed by zones as established in a model description. Only buildings on the main utility distribution line with typical high- and low-bay spaces were analyzed. Table B-I contains a description of the applicable buildings.

The following ground rules were established.

1. The model would be derived from one JSC building.

2. Only buildings on the main utility distribution line would be considered.

3. The building selected would have metered data.

4. The building would have typical high-bay space and low-bay space.

5. The zones of the model selected would reflect typical components of buildings on the site.

Table B-I is a list of the candidate buildings for energy analysis and design evaluation. An analysis of this list showed that building 13 (Structures and Mechanics Laboratory) was typical of approximately 73 percent of the total site floorspace. Therefore, building 13 was selected for analysis. A study[8] provided a description of building 13, its heating, ventilation, and air-conditioning (HVAC) systems, and the zone identification method used for analysis. This building model was analyzed for heating and air-conditioning

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[8] Energy Conservation Study at JSC for Urban Systems Project Office. Prepared by Clovis Heimsath and Associates, January 1976.

loads using the NECAP (NASA Energy-Cost Analysis Program) computer program.[9] The methodology used for programing was as follows.

1. Divide building 13 into 13 different spaces for load calculations and 5 different zones for air-handling systems.

2. Use NECAP thermal load analysis subprogram to calculate building heating and cooling loads, and determine effects of solar radiation, building geometry, weather data, material heat-transfer coefficients, and shading parameters.

3. Use NECAP systems and equipment simulation subprogram to determine fan supply and outside air requirements and fan motor sizes.

The program computes the components of heating and cooling loads on a monthly basis. The load components are roof, walls, window (conduction), window (solar), occupants, lights, equipment, sensible total, latent total, and total. Figure B-1 is an example output of the program. The hourly loads for the maximum annual heating and cooling conditions are shown for a single space in the building 13 model. Table B-II shows the maximum annual heating and cooling load for each space in the building. The maximum cooling load occurs in the high-bay area (space 7). Analysis of energy consumption shows that the high-bay areas constitute 50 to 60 percent of the energy consumption for cooling.

The analysis of heating- and cooling-load components consists primarily of retrofit studies to determine the energy-saving benefits obtained by reducing the magnitude of HVAC loads. The model, building 13, has already been adjusted to reduce energy consumption by reducing lighting levels, changing set points on pretreatment air handlers, and cutting off the heating plant from late April to mid-November. The previously discussed HVAC loads were based on the adjusted model description. These loads were used in retrofit studies to determine viable energy-saving options. Retrofit options for energy conservation that were considered are as follows.

1. Use solar reflective film on windows.

2. Reduce fresh-air ventilation.

3. Shift work schedule, shorten cooling mode hours, lighten heating mode hours.

Personnel at JSC, active in the JSC energy conservation program, have been well aware of the energy savings available through the reduction of the

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[9] W. C. Rochelle; D. K. Liu; R. D. Stallings; and E. S. Riley: Prediction of Building Heating and Cooling Loads and Air-Handling Requirements for NASA-JSC Building 13. Lockheed Electronics Company, Inc., TM-6009, January 1976.

quantity of outside air taken into ventilation systems. There are numerous work areas at JSC, however, in which certain chemical usage or operations prescribe the amount of outside ventilation air. A program of reducing outside ventilation air wherever possible had been accomplished before this study. Because of ongoing tests involving hydraulic fluids, building 13 had not been included in this program.

It was assumed for this study that implementation of the proposed options would occur in 1978. Capital cost is the direct cost for labor and materials for performing the retrofit option projected to 1978. The energy saved is the estimated reduction in the mechanical load (building 24) due to implementing the retrofit option. This quantity is determined by manipulating the data from the loads printout of the computer program (NECAP). Plant efficiency is the thermal efficiency of the JSC mechanical plant for heating and cooling equipment. This value is used to convert heating and cooling loads to energy.

The fuel cost is the cost to the Government of fuel used by the JSC Central Heating and Cooling Plant for heating and cooling. The 1980 projected fuel cost of \$3.08/GJ (\$3.25/10^6 Btu) has been used. Figure B-2 shows the annual component loads for heating and cooling in bar-chart form. Note that the largest single factor affecting heating and cooling loads is mechanical ventilation, which accounts for 51 percent of the total annual cooling load and 75 percent of the total annual heating load. By reducing the quantity of fresh air taken into the system, energy consumption can be reduced considerably. This technique conforms to General Services Administration/Public Building Service (GSA/PBS) guidelines for energy conservation in Federal office buildings.

Currently, the design ventilation rate used is one air change per hour in low-bay (office) zones. In high-bay zones, ventilation rate is determined by the activity being performed in a particular area. These rates exceed American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc. (ASHRAE) requirements for human comfort. If the ventilation rate is reduced to one-half, a cooling-load reduction of 25.5 percent and a heating-load reduction of 37.5 percent could be expected. These reductions represent an annual HVAC energy savings of 27.9 percent.

These ventilation rates are well within the requirements of the Houston Building Code as well as the ASHRAE recommendations considering the occupancy profile for building 13. Figure B-3, based on a contractor study,[8] is a comparison of the current JSC ventilation rate with these sources.

A further energy saving could be realized by varying the rate of fresh-air intake on an hourly basis according to the outside air temperature. For example, during the summer, the ventilation rate could be relatively high in the morning (e.g., 1.5 air changes per hour) but decrease to nearly zero

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[8] Energy Conservation Study at JSC for Urban Systems Project Office. Prepared by Clovis Heimsath and Associates, January 1976.

as the outside air temperature reaches its peak. During the winter, the opposite pattern would be followed.

A means of further reducing fresh-air intake would be by installing an electrostatic or an activated charcoal filtering system into the air-handling system. The Houston Building Code permits a further 50-percent reduction in minimum ventilation rates if this method is used. Both ASHRAE and the GSA/PBS guidelines recommend this approach. It is important to consider, however, that where electrostatic filters have been used at JSC, cleaning and maintenance have been problems. If this option is taken, high priority should be given to selecting low-maintenance equipment. The total energy savings achievable is increased to 42 percent of the total HVAC energy by using approximately 25 percent of current ventilation rates. Figure B-2 shows the relative effect of ventilation on cooling and heating energy as well as the savings gained by these methods to reduce the ventilation rates.

A ventilation rate higher than those discussed may be desirable as various conditions dictate. In laboratory areas where toxic substances are used, higher ventilation rates are necessary. Odor control is another problem, though an activated charcoal filtering system would offset this. Generally, however, the rate of fresh-air intake should never be less than the exhaust rate of toilet rooms and/or fume hoods. Reduction in building pressurization could be a problem in such dust-free areas as computer rooms.

The final limiting factor is human comfort. Though the ventilation rates discussed are higher than the minimum required for human comfort according to ASHRAE tests, a test building at JSC should be used to establish acceptable minimums based on subjective evaluations. The cost of reducing ventilation rates is estimated on the basis of 4 man-hours per fresh-air damper for re-adjusting dampers.

Solar heat gain through windows accounts for 4 percent of the cooling load. However, this heat gain aids the mechanical system during the heating period so that the net annual solar effect is less than 3 percent. This load is a small portion of the total because the generous overhangs, venetian blinds, and high-bay areas with no glass help maintain low solar loads on the overall system.

Commercially available solar reflective film, which is adhered to the glass, can reduce solar loads by 76 percent. This reduction would decrease the total HVAC energy consumption by 2 percent. The current installed cost of this material is $\$12.92/m^2$ ($\$1.20/ft^2$). The payback time for installing this material on all windows is 25.9 years. If it is installed on the east, west, and south exposures only, the payback time is 21.0 years. If it is installed on the east and west exposures only, the payback time is 18.1 years.

The distribution of HVAC energy between seasons shows that considerably more energy is used during the summer than during the winter. Heating and cooling energy during the months of November through April is 36.7 percent of the total annual energy, whereas the air-conditioning energy from May through October is 63.3 percent of the total (i.e., the heating coils are turned off during these months).

It would appear that an energy saving would result from reducing the working hours and the hours of plant operation during the summer months and compensating by increasing the working hours and plant operation hours during the winter. A summer work schedule of 8 a.m. to 4 p.m. (7 hours) and a winter work schedule of 7:30 a.m. to 5:30 p.m. (9 hours), giving an average workday of 8 hours, was assumed. The energy savings, however, were almost negligible because loads eliminated during the summer were during hours of relatively low cooling load, whereas the heating hours added during the winter had relatively high heating loads. One way of avoiding this offsetting effect is to shut down the entire JSC site during the peak cooling month (August) and to increase the workday by 1 hour the rest of the year to compensate. The energy saved is 3.8 percent of the total HVAC load.

In summary, the JSC buildings have already been adjusted to reduce HVAC energy by shutting off nonessential HVAC systems during nonworking hours, reducing lighting levels, changing set points on pretreatment air handlers, and cutting off the building systems from late April to mid-November. The major remaining opportunity for energy saving is in reducing the ventilation rate. A savings between 28 and 42 percent of current HVAC energy consumption could be obtained depending on the effects of reduced ventilation rates on human comfort. Use of electrostatic filters is acceptable under some conditions and the other options assessed are unacceptable for various reasons, as outlined. The results of the analysis are summarized in table B-III.

1. A 50-percent reduction in fresh-air ventilation rate: This reduction appears to be easily achievable within limitations discussed previously. Estimated savings are 27.9 percent of HVAC energy with a small retrofit cost.

2. Electrostatic filters: This method to further reduce the ventilation rate has a 2.3-year payback time based on estimates of capital cost, operation, and maintenance. This method would seem to be acceptable if the low ventilation rate is psychologically acceptable and if physical space exists for the addition of the equipment. Estimated savings are as much as 14.0 percent of HVAC energy.

3. Solar reflective film on windows: Because of existing external and internal shading devices, this option results in excessive payback times. That is, HVAC energy savings are between 1.2 and 2.1 percent with payback times between 18.1 and 25.9 years for various configurations.

4. Summer/winter work schedule shift: A savings of 3.8 percent of HVAC energy can be achieved. It is assumed that this amount does not justify the radical schedule shift required.

TABLE B-I.- BUILDING SURVEY OF JSC

Building number	Gross area, m^2 (ft^2)	Area suitable for analysis, m^2 (ft^2)	Comment
1	18 370.5 (197 739)	18 370.5 (197 739)	
2	5 157.7 (55 517)	761.6 (8 198)	Partial: auditorium
3	1 883.6 (20 275)	0	Cafeteria
4	9 744.5 (104 889)	9 744.5 (104 889)	
5	5 943.1 (63 971)	5 943.1 (63 971)	
7	10 128.8 (109 026)	10 128.8 (109 026)	
8	5 001.6 (53 837)	5 001.6 (53 837)	
9	5 960.4 (64 157)	5 960.4 (64 157)	
10	7 812.6 (84 094)	6 648.7 (71 566)	Partial: Testing lab
11	1 489.9 (16 037)	0	Cafeteria
12	5 824.9 (62 699)	4 706.4 (50 659)	Partial: data center
13	6 360.6 (68 465)	6 360.6 (68 465)	Baseline
14	3 623.6 (39 004)	2 787.5 (30 004)	Partial: anechoic chamber
15	6 765.6 (72 824)	6 765.6 (72 824)	
16	13 877.8 (149 379)	13 877.8 (149 379)	
17	7 824.6 (84 223)	7 824.6 (84 223)	
18	145.7 1 568)	0	Radar
24	3 844.3 (41 380)	0	Utility plant
24A	27.9 (300)	0	Storage
25	590.9 (6 360)	0	Fire station
29	4 486.8 (48 296)	2 757.3 (29 679)	Partial: flight acceleration
30	23 515.0 (253 114)	0	Mission control
31	5 416.7 (58 305)	5 416.7 (58 305)	
32	12 945.3 (139 342)	3 920.5 (42 200)	Partial: simulation chamber
32A	930.5 (10 016)	0	Testing
32J	278.7 (3 000)	0	MIUS testing
32K	33.4 (360)	0	Storage
32M	41.8 (450)	0	Storage
33	1 292.7 (13 915)	1 292.7 (13 915)	
34	383.1 (4 124)	0	Generator
35	2 724.6 (29 327)	2 724.6 (29 327)	
36	5 599.5 (60 273)	5 599.5 (60 273)	
37	7 676.3 (82 627)	7 676.3 (82 627)	
37A	167.8 (1 806)	0	Equipment
37B	5.9 (63)	0	Vaporizer
38	396.3 (4 266)	0	Storage
41	72.6 (782)	0	Health
44	5 208.1 (56 060)	5 208.1 (56 060)	
45	12 625.3 (135 898)	12 625.3 (135 898)	
47	(a)	0	Southwestern Bell
48	1 223.8 (13 173)	0	Utility
49	5 168.0 (55 628)	2 529.6 (27 228)	Partial: not Vibration and Acoustic Laboratories
Totals	210 570.8 (2 266 569)	[b]154 632.3 (1 664 449)	

[a]Not available.

[b]73.4 percent of gross area.

TABLE B-II.- MAXIMUM HEATING AND COOLING LOADS, BUILDING 13

Space	Heating extraction rate, kW (Btu/hr)	Cooling extraction rate, kW (Btu/hr)
1	-43.0 (-146 806)	28.4 (97 031)
2	-28.1 (-95 909)	23.6 (80 549)
3	-34.0 (-116 149)	31.4 (107 124)
4	-28.1 (-95 930)	26.1 (89 019)
5	0	23.7 (80 915)
6	-7.2 (-24 583)	30.8 (105 173)
7	-101.4 (-346 235)	153.9 (525 570)
8	-7.3 (-24 970)	19.3 (65 844)
9	-42.9 (-146 673)	30.2 (103 088)
10	-28.8 (-98 261)	24.6 (84 011)
11	-34.8 (-118 716)	33.1 (113 135)
12	-28.7 (-98 083)	28.5 (97 233)
13	-1.4 (-4 847)	30.7 (104 702)

TABLE B-III.- SUMMARY OF LOAD REDUCTION OPTIONS, BUILDING 13

Option	Annual energy saved, GJ (Btu)	Total HVAC saved, percent	Annual cost of energy saved, dollars	Estimated capital cost for option, dollars	Estimated payback time, yr
Ventilation					
A - 50 percent reduction in vent[a]	49 952 (47 377×10^6)	27.9	153 845	6 000	0.04
B - Add electrostatic filters	24 955 (23 669)	14.0	76 925	180 000	2.3
C - Both A and B[a]	74 865 (71 006)	41.9	230 770	186 000	.8
Window solar					
D - Solar film on all windows	3 710 (3 519)	2.1	11 435	296 000	25.9
E - East, west, and south windows only	3 119 (2 958)	1.7	9 614	202 000	21.0
F - East and west windows only	2 227 (2 112)	1.2	6 864	124 000	18.1
Work schedule shift					
G - Shift 1 hr from summer to winter	1 161 (1 101)	.6	3 578	0	0
H - Increase workday 1 hr with Aug. shutdown	6 787 (6 437)	3.8	20 920	0	0

[a]Items recommended for further consideration.

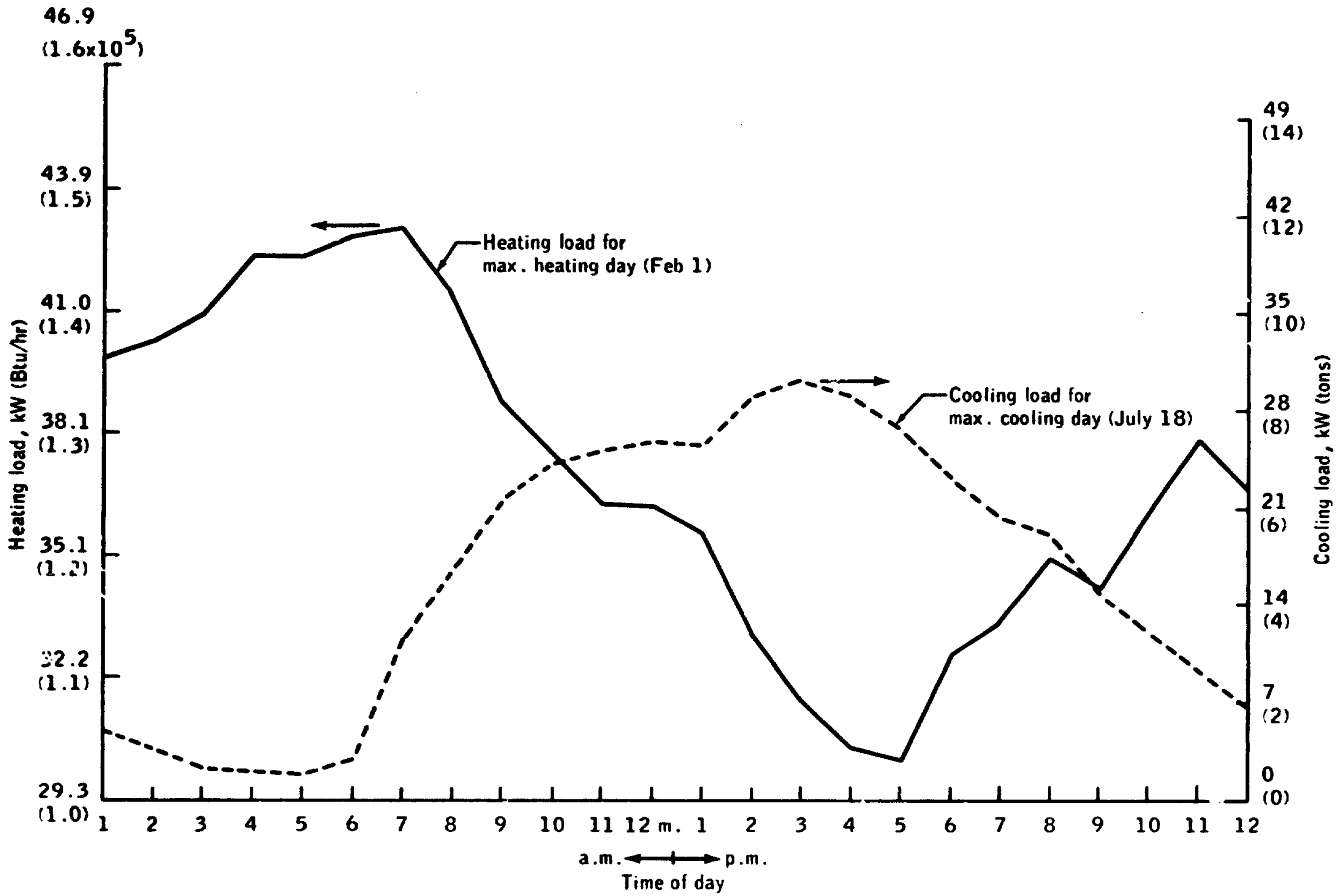


Figure B-1.- Heating- and cooling-load profiles for space 1 in building 13.

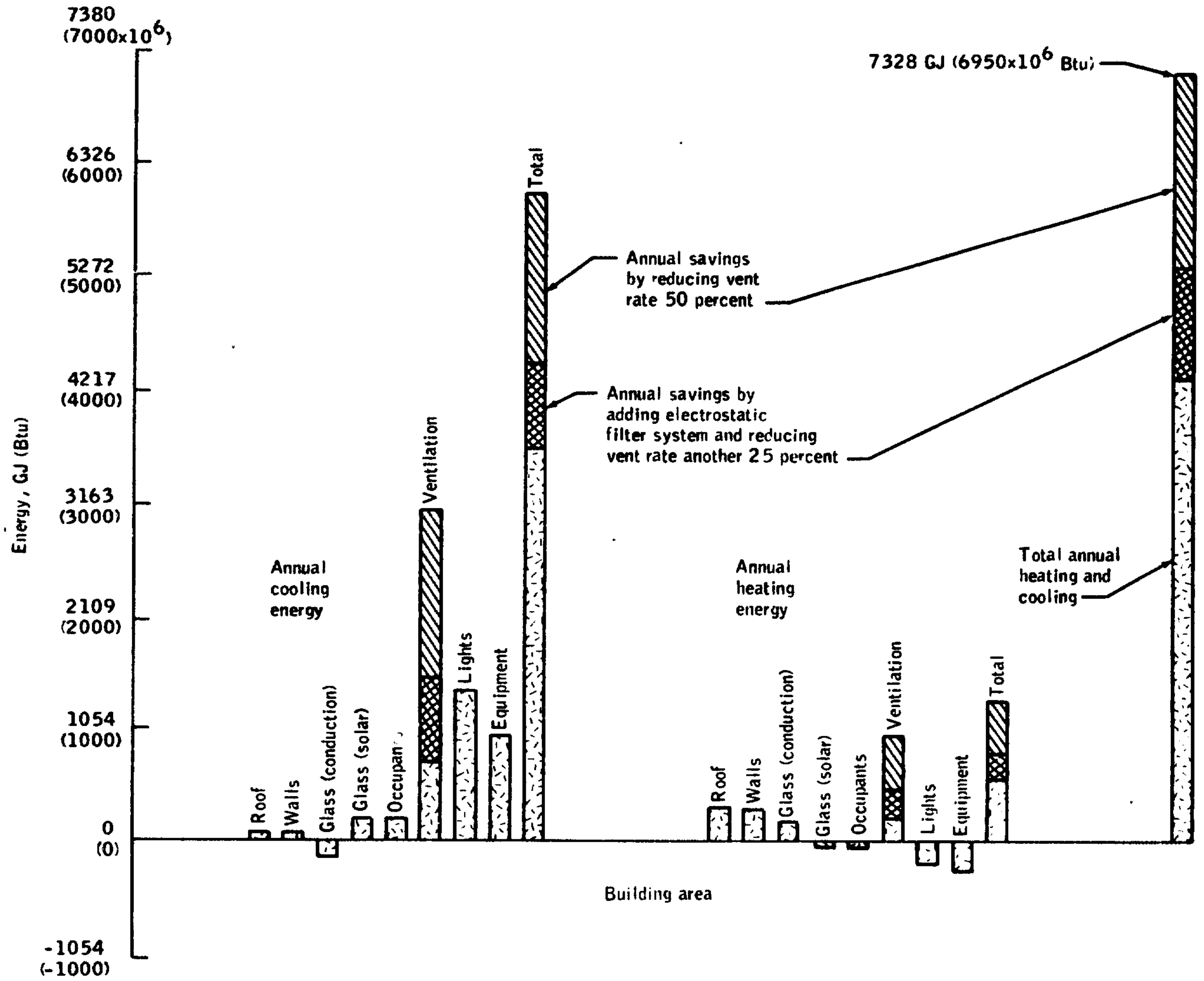
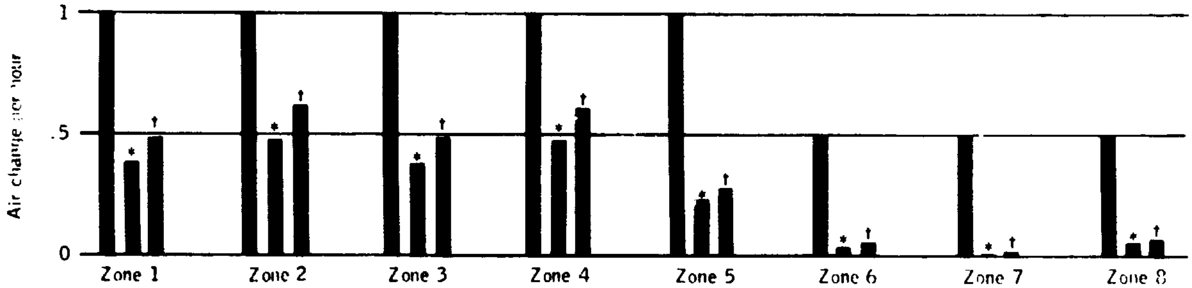
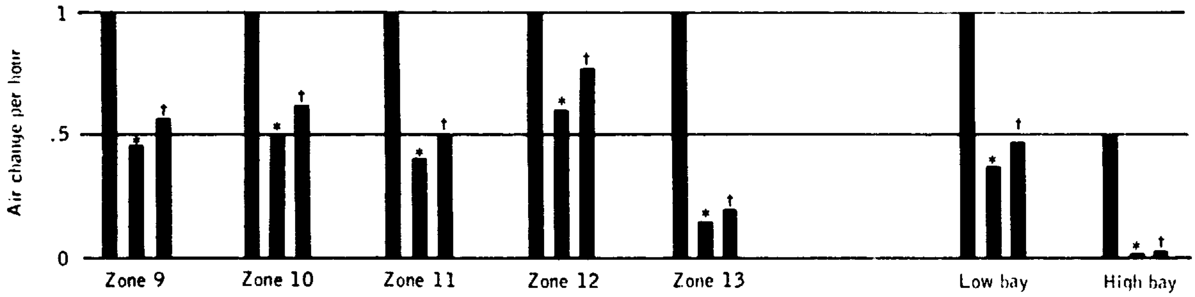


Figure B-2.- Effect of ventilation-rate reduction on building 13 HVAC energy consumption.



(a) Zones 1 to 8.



(b) Zones 9 to 13, low bay, and high bay.

*Houston Building Code requirements for offices in which smoking is permitted: 0.0057 m^3/sec (12ft^3/min) fresh air per person; quantities may be reduced 50 percent if electrostatic filtering is provided.

† ASHRAE - recommended fresh air for office in which smoking is permitted: 0.0071 m^3/sec (15 ft^3/min) per person; quantities may be reduced if air filtering and/or odor control is provided.

Figure B-3.- Fresh-air ventilation rates for building 13. For each zone, left-hand bar represents JSC current ventilation rate, center bar represents Houston Building Code minimum ventilation-rate requirement, and right-hand bar represents ASHRAE-recommended ventilation rate.