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SOLAR ENERGY SYSTEM DESIGN FOR A LOBSTER AQUACULTURE FACILITY

Final Report

September 30, 1977

Work Performed Under Contract No. EG-77-C-02-4274

Sanders Associates, Inc.
Defensive Systems Division
Nashua, New Hampshire

MASTER

U.S. Department of Energy

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Solar Energy

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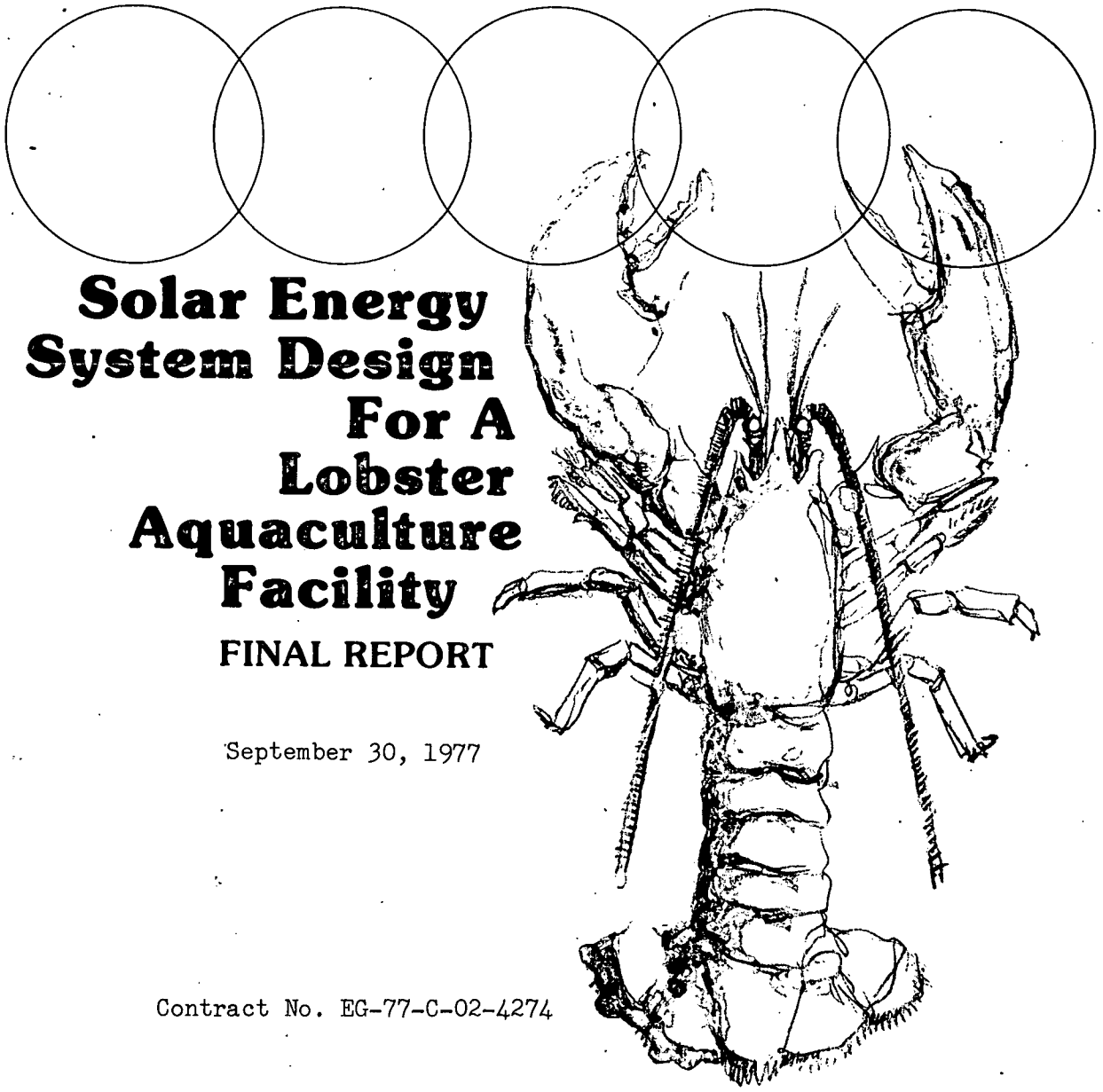

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ABSTRACT

In aquaculture, as in most manufacturing processes, the operating cost is greatly dependent upon the cost of energy. The objectives of Contract E6-77-C-02-4274 were to: (a) analyze the power requirements for a lobster aquaculture plant currently under development by Sanders Associates, and (b) to evaluate the use of solar energy as a cost reduction measure in plant operation.

A flat plate collector system capable of supplying heat alone was compared with a total energy system in which both electrical power and heat were supplied. The flat plate collector was not cost effective because when heat was needed in December, the least amount of heat was available from solar radiation. Therefore, the collector area and cost were prohibitive. However, the total energy system was cost effective when the capital investment was amortized over ten or more years.

The optimum solar power plant was designed to provide 100% of the average yearly power demands, or 60% of the December power requirement. This plant would consist of 60,000 square feet of mirror surface (3.5 acres of land for 40% packing density) which would concentrate 1500 to 2000 suns on a receiver mounted on an 85 foot tower. In the tower would be the three storage stoves which would contain the heat required to operate a 343 KWe Brayton gas turbine engine and alternator for 27 hours.

Equipment to generate 3 million kw-hr annually will cost an estimated $1.3 million. When amortized over 20 years, the cost would be $0.01/kw-hr, compared with $0.02/kw-hr for commercial electric power plus oil heat. This analysis assumed an inflation rate and interest rate of 7.5% per year, maintenance cost of 5% of capital investment per year. and fuel inflation rate of 10% per year.

The recommended solar powered aquaculture plant, using 130,000 square feet of factory space, will produce 800,000 market-size lobsters per year. To heat and operate the production facility, 343 kw of electric power and a maximum of 885 kw of heat must be supplied continuously by the power plant.

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SECTION 1
INTRODUCTION AND SUMMARY

1.1 INTRODUCTION

Solar energy has a key role in the animal husbandry of marine species. Sanders Associates owns and operates a facility for the aquaculture of North American Lobsters. This pilot plant for the growth of North American lobsters offers Sanders a convenient data source for assessing the role of solar energy in powering a commercial lobster aquaculture plant. This study contract (ERDA Contract No. E5-77-C-02-4274) was directed toward: (a) defining equipment for evaluation at Sanders' pilot plant located in Kittery, Maine, and (b) assessing cost for application to a full-scale commercial plant.

1.2 SUMMARY

The energy requirements of a commercial aquaculture plant were projected from the Sanders operated and owned pilot plant. A commercial plant would be capable of marketing 800,000 lobsters per year in 130,000 square feet of factory space. Most of the energy would be used by the commercial plant to: (a) circulate nutrient-rich seawater through the plant, and (b) adequately aerate the seawater in the plant. The energy expended in pumping the water would heat the seawater temperature to 60°F, the optimum for rapid growth of lobster. Energy would also be required for light and heat during the winter months. In summary, 343 KWe would be required for electric power and 512 KWt would be needed for heat. Assuming that: (a) the 512 KWt would be recovered from the power generator exhaust, and (b) all electrical power would be converted to heat, a total energy plant operating at a thermal conversion efficiency equal to or less than

$$\frac{343}{855} = 0.4$$

would meet all the energy requirements of the facility.

Flat plate collectors were evaluated for the total energy plant application. However, even when using the most efficient, tubular evacuated collectors and an optimum organic Rankine engine, the overall efficiency would be less than 0.07 and cost effective operation would not be achievable. Therefore, a more efficient power conversion cycle was recommended.

The recommended concept consists of a hot air solar receiver and a low pressure ratio Brayton gas turbine to generate electric power. This system would have an overall efficiency of 0.275. This efficiency could be increased to 0.316 by adding the organic Rankine engine as a bottoming cycle.

The total energy power plant (Figure 1-1) would be located adjacent to the aquaculture facility on 3.5 acres of cleared land. An 85 foot tower on the south side of the field would contain:

- The solar receiver, into which solar radiation is projected by 60,000 square feet of mirror surface.
- Three sensible heat storage stoves
- The Brayton and Rankine gas turbines and generators
- Fossil fuel combustor
- Controls

Power lines and a hot air duct would connect the power plant to the production facility.

The block diagram (Figure 1-2) shows the aquaculture system supplied with temperature-controlled make-up water from the ocean. This water is continuously circulated, filtered and aerated before being returned to the ocean through a heat exchanger which preheats

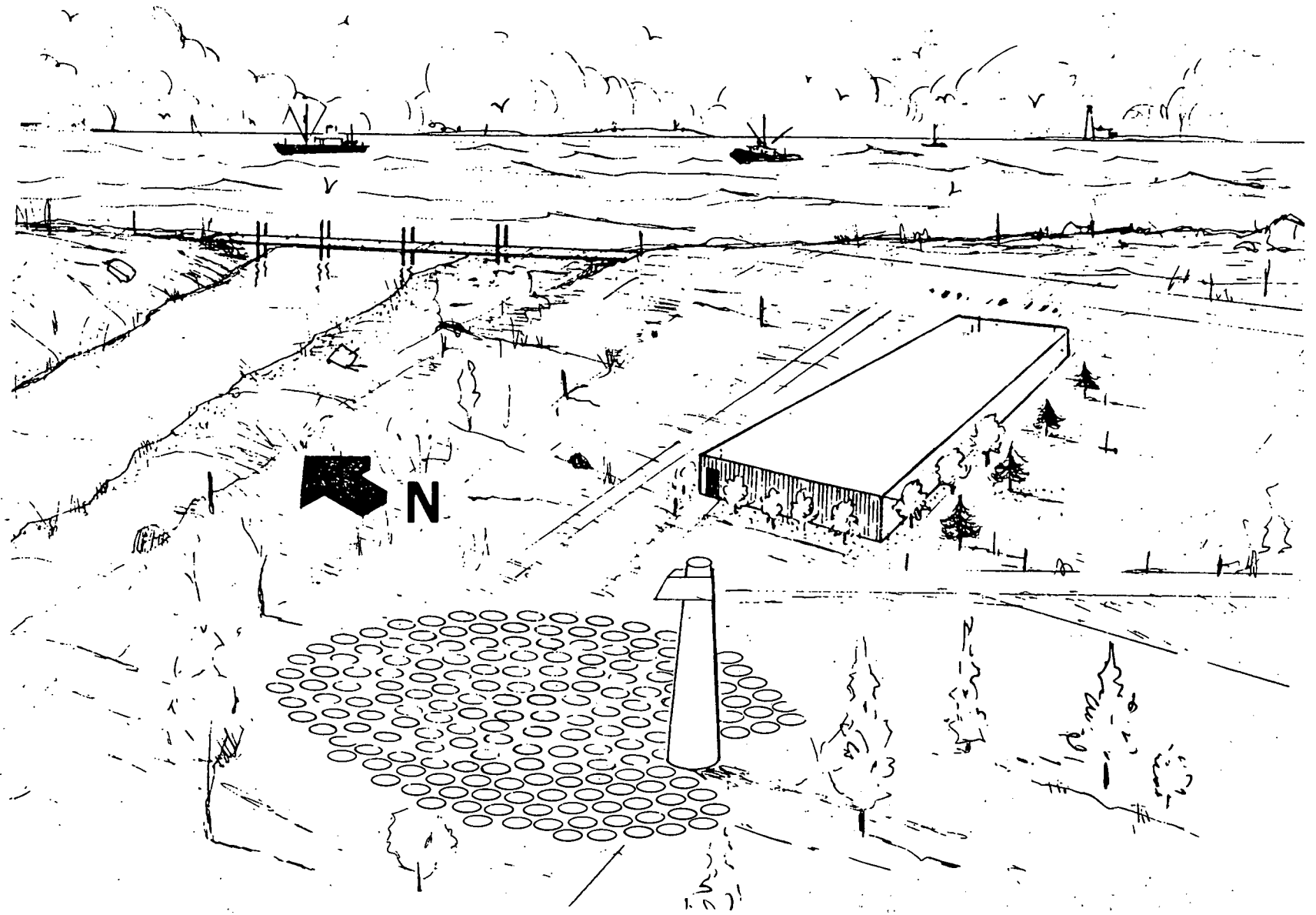

Figure 1-1. Artist's Concept - A Total Energy Aquaculture Plant

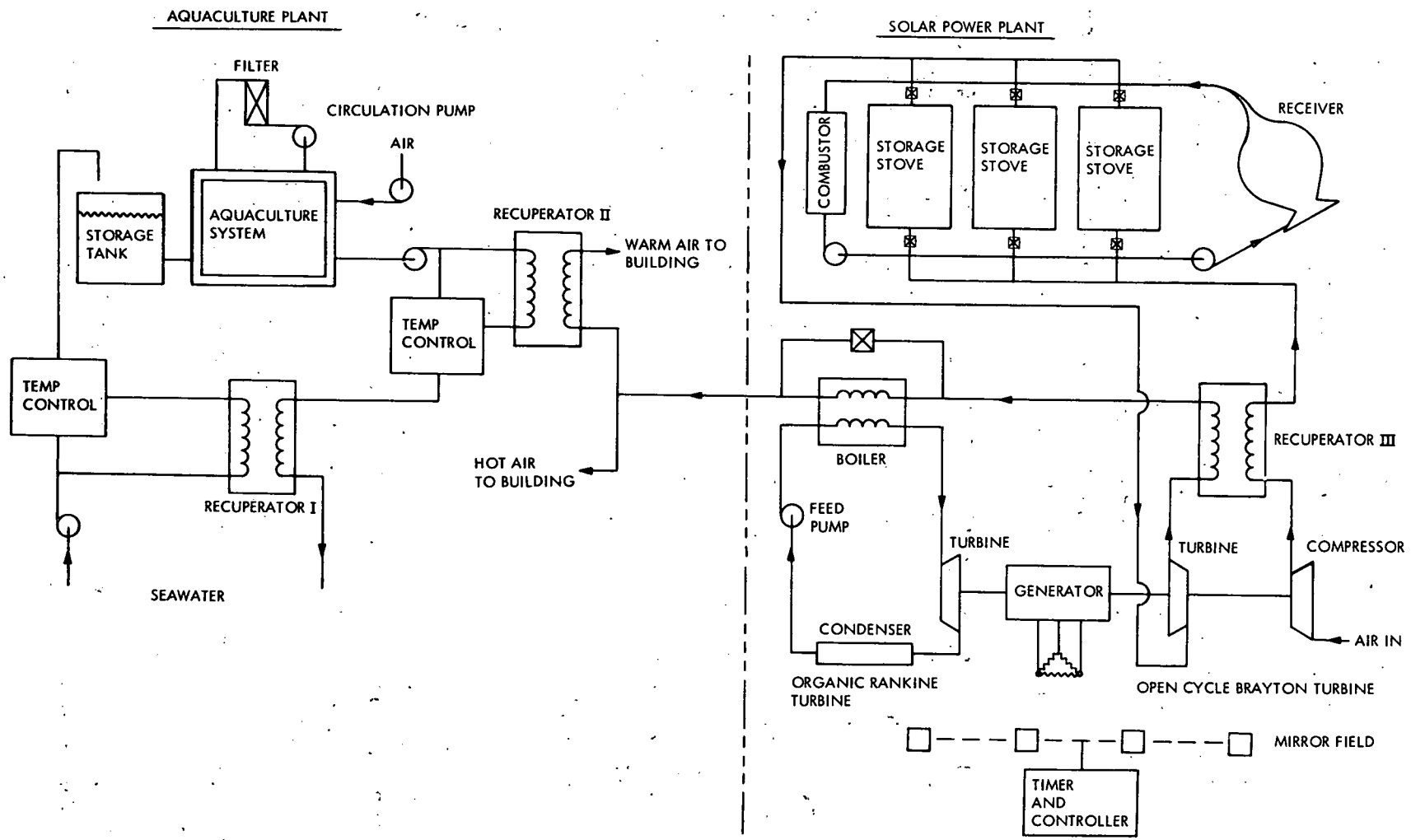

Figure 1-2. Total Energy System - Block Diagram

the incoming water. Electric power is continuously generated by the gas turbine.

A solar receiver collects solar radiation and three checker stoves store the energy as heat. Each stove would be capable of operating the 343 KWe gas turbine generator for nine hours. Should solar radiation be inadequate to charge a stove, additional heat could be supplied to the stove by a separate, fossil fuel combustor.

The power cycle would be totally independent of the charging cycle. Using valves, the gas turbine would be connected to any fully charged stove, such that the compressed air leaving the compressor would be heated by recuperator III to 1200°F prior to further heating to 2000°F in passing through the checker stove. The hot compressed air would then expand through the turbine to generate power. Heat would be recovered from the exhaust gas by passing the gas through recuperator III. The 400°F exhaust from the recuperator would then pass into the boiler of the organic Rankine cycle turbine. The gas would emerge from the turbine at 200°F and would be either exhausted from the plant or used to heat the aquaculture building.

The mirror field has been designed to collect solar radiation to supply 60% of the December energy needs based on meteorological data for Portsmouth, NH. The stoves were sized to store heat for 27 hours of gas turbine operation. Therefore, the stoves would be capable of storing all the solar energy collected by the receiver except during the summer months when excess energy is available. During December, and during extended periods of overcast, the required energy would be provided by an alternate source such as a fossil fuel combustor.

This total energy plant would cost approximately 1.3 million dollars. When amortized over 30 years, allowing 5% per year for

maintenance, 7.5% interest rate, 7.5% inflation rate and 10% fuel inflation rate, the equivalent power cost in 1977 dollars would be approximately $.009/kw-hr. This would compare with $0.030/kw-hr which is the present cost of electrical power and electric heat. Included in the $.009/kw-hr is $.002/kw-hr for fossil fuel expended when solar radiation is inadequate.

Since the total energy plant would be self-contained, no power plant demand charge would be paid. The plant could be located on a coastal island where land is inexpensive, seawater is uncontaminated and electric power is not presently available. A total energy system is recommended for further engineering evaluation of its application to lobster aquaculture and other processing plants having similar electrical and thermal needs.

SECTION 2
HEAT AND POWER REQUIREMENTS

2.1 HEAT REQUIREMENTS

In the aquaculture application, some purely thermal energy is necessary to maintain the temperature of both the process water and the building. Flat plate solar collectors are the obvious candidate to supply this thermal requirement.

2.1.1 Building Heat Load

The building heat losses were calculated for the commercial size building (see Appendix A and Figure 2-1) using the formulas of Table 2-1. Results are presented for both average and maximum environmental design conditions in Table 2-2.

Included in the calculation is an increment due to high moisture content (100% humidity). A high value was assumed because in an aquaculture facility all the tanks are open at the top and actual equilibrium humidity condition will be nearly 100% and therefore this calculation provides an upper limit.

In view of the ample opportunity for equilibration of the aquaculture water with the building atmosphere, no special provision has been made to heat the building beyond that heat delivered by the water, the aeration process and the waste heat of the many pump motors. Therefore, no radiators or heater are needed to maintain building temperature, all the building heat loss will be supplied by adding heat to the process water.

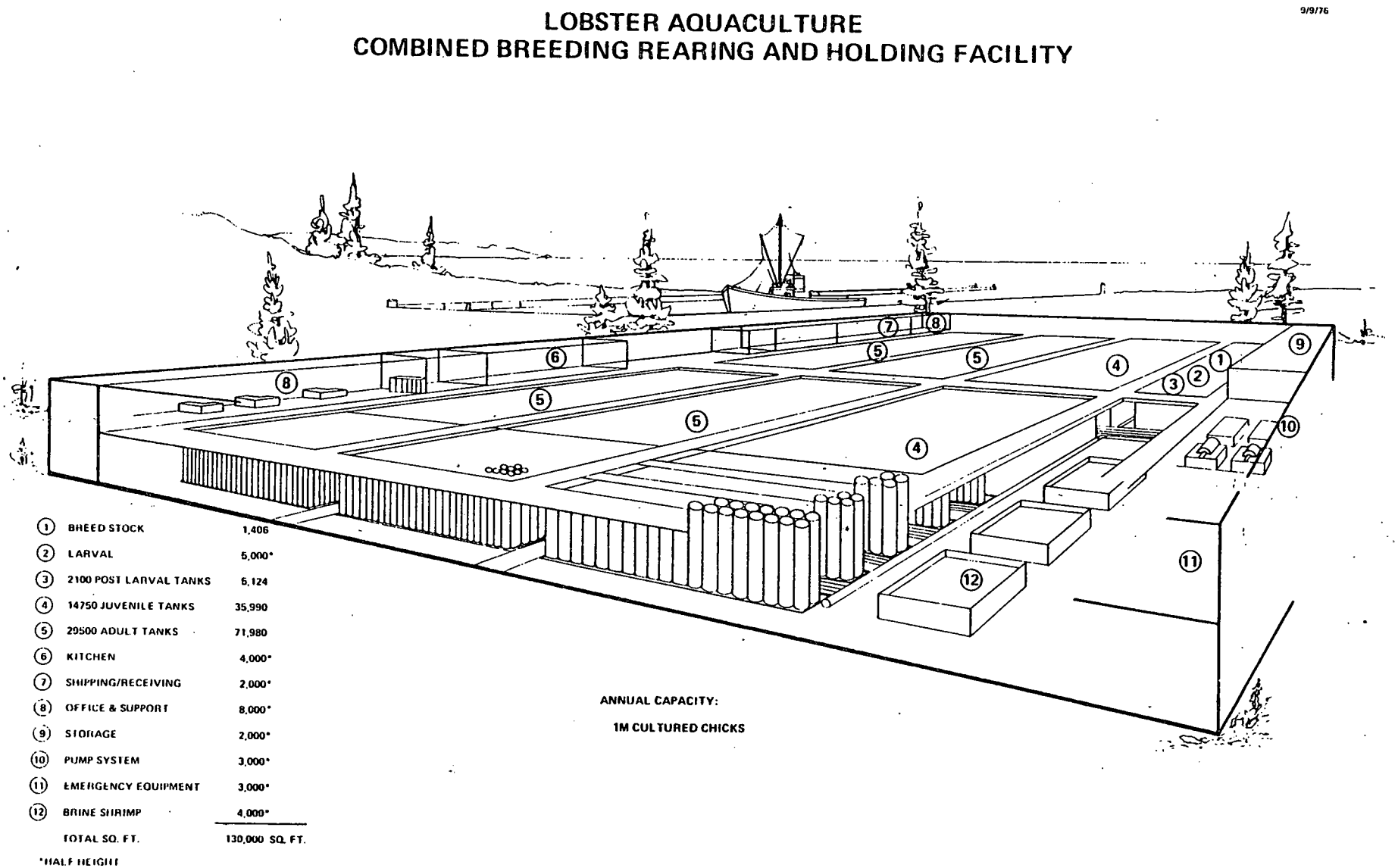

Figure 2-1. Lobster Aquaculture Combined Breeding, Rearing and Holding Facility.

TABLE 2-1

HEAT LOSS CALCULATION FORMULAS

Building Dimensions are:

length = 650 ft

width = 200 ft

height = 30 ft

Building Heat Loss ($\dot{Q}_B$) is

$$\dot{Q}_B = SU(T-T_{AMB}) + FP + X \quad V + 0.24(T-T_{AMB}) + \Delta h_{AIR}$$

where

F = a perimeter loss factor, taken from the ASHRAE HANDBOOK

= 21 for $T-T_{AMB} < 30^{o}F$

= 34 for $T-T_{AMB} > 30^{o}F$

X = fractional change of air volume <u>per hour</u> due to infiltration, taken as 0.1 changes per hour for the projected plant design. Tighter building design could materially reduce the building heat loss, but is beyond common building practice.

Δh_{AIR} = heat required to evaporate the moisture contained in 1 pound of air at temperature T,

$$= 16.7 \text{ Btu/lb} = 1054 \frac{\text{Btu}}{\#H_2O} \times 0.016 \frac{\#H_2O}{\#AIR}$$

TABLE 2-2

BUILDING HEAT LOSSES

BUILDING	CONDITIONS	SURFACE LOSS (Btu/Hr)	PERIMETER LOSS (Btu/Hr)	DRY AIR INFIL-TRATION LOSS (Btu/Hr)	SATURA-TED AIR INFILTRA-TION LOSS INCREMENT (Btu/Hr)	TOTAL (Btu/Hr)
Commercial Plant	AVERAGE $\Delta T = 20^{o}F$ $X = 0.1$ $F = 21$	218,000	35,700	181,000	605,700	1,041,000
Commercial Plant	DESIGN MAX. $\Delta T = 65^{o}$ $X = 0.1$ $F = 34$	709,000	57,800	589,000	605,700	1,962,000

2.1.2 Process Water Heat

The 24,300,000 pounds (1.1×10^7 kg) of water is changed on the average once in seven days to prevent excessive accumulation of waste products. A substantial amount of energy is required to heat the incoming seawater by the required average of 20^oF (11.1^oC) to reach the 65^oF (18.3^oC) design temperature.

A large part of this energy is saved by using a heat exchanger made of titanium with a cleanable structure to allow control of fouling. At the given flow conditions (mass flow rate both in and out at 145,000 lb/hr), with a "hot" outlet temperature of 65^oF and a "cold" inlet temperature of 45^oF (average), a heat exchanger effectiveness of 80% is anticipated which means that 80% of the energy that would otherwise be required to raise the entire mass of water by 20^oF is supplied by the recuperator. A tabulation of heat exchanger performance over the months of the year is provided in Table 2-3. Beyond the energy retained in the system as returned by the heat exchanger, additional energy comes from the mechanical energy dissipated in the water by the filtration process, and from the aeration system.

2.1.3 System Time Constant

A unique feature of this aquaculture system is the large time constant for temperature change due to the exceptionally large mass of water involved. This time constant, and the thermal averaging which it implies, is considerably longer than the storage time of any thermal storage that would be considered for use with any normal building heating application. Therefore, the time constant is a major factor in plant design. It materially reduces the standby heating requirements which are essential for the present plant.

TABLE 2-3

HEAT EXCHANGER PERFORMANCE $\eta = 0.80$

MONTH	T_{WATER} (cold side, In)		T_{WATER} (cold side, Out)	
	(^oF)	(^oC)	(^oF)	(^oC)
JAN	35.9	2.2	59.2	15.1
FEB	34.6	1.4	58.9	14.9
MAR	36.7	2.6	59.3	15.2
APR	42.4	5.8	60.5	15.8
MAY	48.8	9.3	61.8	16.6
JUN	55.0	12.8	63.0	17.2
JUL	59.5	15.3	63.9	17.7
AUG	61.3	16.3	64.3	17.9
SEP	58.2	14.6	63.6	17.6
OCT	52.6	11.4	62.5	16.9
NOV	47.2	8.4	61.4	16.3
DEC	40.3	4.6	60.1	15.6

If τ is the time constant; i.e., time required for the temperature to drop to 1/e of its original value, and τ_8 is the time to drop the allowed 8^oF from the original $\Delta T = 20^oF$, then the numerical values are as given below:

$$\tau = \frac{Mc_p\Delta T}{\dot{Q}} = \frac{M}{\dot{M}(1-E) + \frac{\dot{Q}}{c_p\Delta T}}$$

$$\tau_8 = \tau \ln \left(1 - \frac{8}{\Delta T}\right)$$

where

M = total mass of water
ΔT = temperature drop
$\dot{Q}$ = heat flow rate
$\dot{M}$ = mass flow rate
E = heat exchanger effectiveness
c_p = 1

For example, with $M/\dot{M} = 168$, recuperator effectiveness (E) (which = 0.80 for the present design), and negligible losses ($\Delta\dot{Q}$) other than the outgoing water,

$$\tau_8 = -\frac{168}{(1 - 0.8)} \ln \left(1 - \frac{8}{20}\right) \quad \text{(hours)}$$

$$= 429 \text{ hrs}$$

A complete shutoff of power would increase $\Delta\dot{Q}$ to the full building loss, decreasing τ_8, but a stoppage of the main circulation would effectively eliminate recuperator losses thereby raising τ_8. The plant could function without circulation power for some time if aeration is maintained. With a standby aeration system, this plant is highly resistant to power failures.

2.2 MECHANICAL REQUIREMENTS

2.2.1 Filtration

The energy required for mechanical functions dominates the power requirement of this aquaculture plant. These requirements are necessitated by the biological needs of the living organisms in the aquaculture plant. Experience has shown that, on the average, one full change of seawater in seven days is adequate to prevent undue buildup of chemical waste. However, the water must be physically filtered once every 2.2 hours. Maintenance of this high flow rate against the resistance of a filtration system consumes substantially more energy than the renewal of seawater.

2.2.2 Aeration

The aeration of the water in each tank consumes even more energy than the filtering process. Each tank is 12-15 ft high (3.7 - 4.6 m), producing a large back-pressure against which a substantial volume of air must be forced. This operation requires more energy than any other item in the plant and is necessary to keep the lobsters alive.

The aeration energy requirements are minimized by the following:

a. Introduction of air at various levels within each tank, so that the average back-pressure is approximately half what it would

be if all air were introduced at the bottom. This measure, however, requires a multiple air pump system with pump lines at the correct pressure for each level of air introduction.

b. The porous plug material through which bubbles are introduced into the tanks must be optimized. Material with an excessively fine pore results in excessive back-pressure and consequent pump energy consumption, whereas material with excessively coarse pore causes excessively large bubbles with unsatisfactory oxygen absorption by the water. Suitable materials have been found through experimentation.

2.2.3 Mechanical Energy Requirements

The mechanical energy requirements of the full scale commercial plant, extrapolated from the pilot plant, total 287 KWe, neglecting pump inefficiencies (see Table 2-4). Aeration and recirculation have the largest energy demand.

2.3 CONSERVATION MEASURES

2.3.1 Building Heat Conservation

The building heat losses must be minimized with building insulation. For the present plants, insulation with a heat transmission factor (U) of 0.05 Btu/hr-ft^2-^{o}F was selected as a cost effective level. Any thicker insulation would impact the structural design. With modular, industrial, prefabricated designs presently being considered for their economic advantages (such as Butler buildings), any insulation features requiring custom design would be exceptionally costly. Other heat losses, not controllable by insulation, are especially significant in an aquaculture plant. Heat loss due to infiltration, which is negligible in small dwellings of high surface/volume ratio, is comparable to the surface heat loss (through the insulation) for the commercial size plant (see Table 2-2).

TABLE 2-4

MECHANICAL ENERGY REQUIREMENTS - COMMERCIAL PLANT

	Energy (Btu/hr)
Main Plant Pump	
145,000 lb/hr flow, seawater inlet with 32 ft head plus 25 psi back-pressure, to include sand filter, heat exchanger, piping	16,600
In-Plant Water Distribution	
pump 145,000 lb/hr through a representative pipe network	102
recirculate water in aquaculture tanks at 1,120,000 lb/hr through an equivalent 25 ft head	360,000
Aeration	
Supply 7400 CFM @ 7 psi average pressure at various levels in aquaculture tanks	576,000
Electricity for Lighting	
Monitoring Instruments, Room Air Circulation, etc.	23,800
Total	979,000 or (287 kw)

Several times as great, and peculiar to aquaculture plants is the heat loss due to moisture infiltration involving the latent heat of evaporated water when it exits the facility in exchange for outside air (presumably of negligible moisture content). This applies when outdoor temperatures are low, even at high relative outdoor humidity. An adequate heat loss estimate was calculated as an upper limit by assuming: (a) zero (negligible) moisture in outside air, and (b) full saturation, or 100% humidity in indoor plant air.

At the assumed low infiltration rate of 10 percent of the building volume exchanged per hour, assumption (b) is considered well justified if no attempt is made to reduce the exposed water surface area. Assumption (a) applies especially in winter when all other heat losses are maximized. Results of calculation are presented in Table 2-5.

2.3.2 Aquaculture System Conservation

Maintenance of the aquaculture water temperature is burdened primarily by the requirement for exchange with fresh seawater, as discussed in Section 2.1. Much of the thermal energy required to heat this water can be recovered from water leaving. A recuperator is an order of magnitude more cost effective than any alternative means of generating the equivalent energy.

Because lobsters are highly sensitive to foreign materials in contact with the water, especially copper, extreme caution is required in selecting a heat exchanger. A particular family of exchangers was selected: The P-600 series by American Standard, referred to as the "Plateflow" type. This series has three features of special value in the aquaculture application:

TABLE 2-5

AVERAGE NET ENERGY REQUIREMENTS

Full Scale Commercial Plant	
Mechanical Energy Requirements	
979,000 Btu/hr (287 kw) (at 100% Eff)	
1,165,000 Btu/hr (343 kw) (at 84% Eff)	
Gross Thermal Requirements	
Building Heat:	1,041,000 Btu/hr
Process Water:	2,940,000 Btu/hr
Total Thermal:	3,981,000 Btu/hr
Deductions from Thermal Requirements	
Mechanical Energy x 1.19 (allows for 84% efficiencies at $\frac{1}{0.84} = 1.19$) (1,165.000) Btu/hr	
Heat Exchanger Contributions at E = 0.80	(2,352,000)
Total Deduction:	(3,517,000)
Net Thermal Requirement	
	464,000 Btu/hr (136 kw)
Total Energy Requirement	1,629,000 Btu/hr (477 kw)

- It is available in a variety of metals, including titanium which is believed to be resistant to corrosive effects of seawater and is harmless to lobsters.

- It is readily disassembled, with full access to all surfaces for cleaning, to remove any fouling that may result from seawater at temperatures highly favorable to growth of micro-organisms.

- The basic construction is highly flexible through the configuration of gaskets between the plates which control the fluid flow pattern through the exchanger. The manufacturer uses a computer-aided design to select the configuration that best suits the combination of hydraulic flow and thermal requirements. Thus, for example, pressure drop can be optimized at the required mass flow rates along with attainment of the maximum heat exchange for the total plate surface area.

2.4 NET ENERGY REQUIREMENTS

In examining the plant's net energy requirements, two items should be considered:

(a) All the mechanical energy used in pumping air or water will also be converted to heat and will contribute to the building heat or process water heat. In addition, any inefficiencies in motors, pumps, etc. (efficiencies might commonly average 85%), will appear as additional heat.

(b) The overall ratio of mechanical to total energy required for the plant is characteristic of the mechanical/total energy ratio of well designed power plants. Table 2-5 shows the average net energy requirements for a full scale plant.

SECTION 3
SYSTEM DESIGN

3.1 INTRODUCTION

The heat and power requirements described in Section 2 are average values which have been converted to peak values for design purposes in Table 3-1. Assuming that all the electric pumps are located within the building and, therefore, contribute to the total thermal requirements, the total thermal energy requirement is 855 kw. This assumes a saltwater heat exchanger which recovers about 80% of the heat from the effluent water. This thermal energy requirement varies with the seasons and weather to the point that cooling is required in the summer months. Cooling can be implemented by bypassing the saltwater heat exchanger.

Mechanical power requirements are constant throughout the year. This power can be either purchased from the Public Service Electric Utility plant or generated locally. Cost of equipment and power will determine the optimum system design.

3.2 FLAT PLATE SOLAR COLLECTORS

3.2.1 Performance and Cost of Commercial Equipment

Flat plate solar collectors may be categorized into three basic groups:

- Inexpensive: <\$5.00/ft^2 uninstalled
- Average flat plate collector, the basis of the current solar market: \$10-\$13/ft^2 uninstalled
- Expensive, new, evacuated tubular solar collectors: \$20/ft^2 uninstalled

The price of collectors is a reflection of state of technology, material composition and, most importantly, rate of heat extraction.

TABLE 3-1

PEAK DESIGN ENERGY REQUIREMENTS

FULL SCALE COMMERCIAL PLANT

MECHANICAL ENERGY REQUIREMENTS

979,000 Btu/hr (287 kw)(@100% Eff)

1,165,000 Btu/hr (343 kw)(@84% Eff)

THERMAL REQUIREMENTS - GROSS

Building heat	1,962,000 Btu/hr (max)
Process water	4,785,000 Btu/hr (max)
Total thermal	6,747,000 Btu/hr (max)

Deductions from gross thermal requirements

mechanical energy x 1.19 (allows for 84% efficiencies at $\frac{1}{0.84} = 1.19$)

(1,165,000) Btu/hr (343 kw)

Heat Exchanger contributions at E = 0.80

(3,828,000) Btu/hr

Total Deduction	(4,993,000) Btu/hr
Net Thermal Requirement	1,754,000 Btu/hr (514 kw)

TOTAL EXTERNAL ENERGY REQUIREMENT

2,919,000 Btu/hr (855 kw)

Ratio, Mechanical/Total 0.40

3.2.1.1 Inexpensive Collectors

The inexpensive collectors generally have an absorber plate of extruded plastic (CPVC or an elastomer such as EPDM are common). To enhance its thermal capabilities, this plastic absorber plate, may be painted with a selective coating of high absorptivity and low emissivity. Choice of acrylic or glass cover system is usually limited to one cover only since the maximum stable temperature of the plastic absorber material is in the 300 - 400°F range. Insulation is meager if not non-existent among the inexpensive collectors due to low re-radiation properties of the materials involved. The application of inexpensive collectors is at this time limited to water heating. Their average 4°F ΔT rise per pass is not sufficient for peak demand space heating during December.

The principle advantages of the inexpensive collectors are:

- Price
- The long lengths in which they are extruded (up to 125 ft) vastly diminishes their plumbing requirements.
- Ease of installation.

Basically, the only problem with the inexpensive collectors (for this application) is whether or not they can withstand summer no-flow conditions.

3.2.1.2 Average Collector

The average flat plate collector which is the basis of the present solar market, has an absorber plate of metal with some type of selective coating, one or two glass or acrylic covers, and some insulation. The absorber plate may have any one of a number of configurations; among these are tube in sheet (Roll Bond), serpertine, tube and fin, or corrugated.

In general, the "average" solar collectors have a ΔT rise of $10^{o}F$ or more per pass and have been shown to be adequate for space heating as well as water heating. Economically, they compare favorably with electric heat and, with increased production and further cost reduction their cost will be comparable to fossil fuel heating costs.

A prime consideration in choosing one of the "average" solar collector systems is that they use off-the-shelf plumbing supplies, thus eliminating undue delays in installation and repair. Unlike the inexpensive collectors, the no-flow condition presents no problem with the "average" collector systems since the maximum stable temperature of the materials employed is well above the stagnation temperature range.

3.2.1.3 Evacuated Tubular Collector

The evacuated tubular collector technology developed over the past few years is still in a research and test installation stage. The current price of these collectors, $\$20.00/ft^2$ uninstalled, makes them economically prohibitive. However, due to their much higher efficiency and output temperatures, research is proceeding in hope of lowering their cost.

The evacuated tubular collectors consist of a black, hollow metal rod usually of copper enclosed within two layers of glass separated by a vacuum. Reflectors, which increase the incident insolation on the absorber tubes, are a recent improvement and virtually give them a $360^{o}F$ absorption area.

There are many advantages of the tubular evacuated collectors. Wind no longer constitutes a large convection loss since the absorber operates within a vacuum barrier. Efficiency is as high as, if not higher than, the best flat plates, and yet does not decrease with low ambient temperatures. Endurance of high temperatures and no flow conditions is excellent. A ΔT rise of $25^{o}F$ per pass is common with

an overall daily rise of 120°F not unusual even for blustery New England winter days. The evacuated tubular collectors, are thus, not only able to supply heat and hot water, but are also able to operate organic Rankine power plants.

In summary, all three types of solar collectors, the "inexpensive" plastic, "average" metal, and evacuated tubular, have their advantages and disadvantages. The answer to which collector system is the optimum for a particular situation is a combination of collector application, whether it be hot water, heat, air conditioning or a combination thereof, and the efficiency of the system (Btu/$) which includes the cost. Table 3-2 gives an overview of solar collectors currently on the market, their cost per square foot, material composition, application, etc.

3.2.1.4 Flat Plate Analysis

Various types of flat plate collectors, their efficiency curves and performance data are compared in Figures 3-1 and 3-2. Owens-Illinois "Sunpack" has the highest efficiency curve when measured by NBS standard test procedures. It also rates the highest whether based on Btu/ft^2 of $/ft^2.

Meeting the thermal load of the aquaculture production plant in December is possible only when solar heating panel area exceeds the roof area. The roof area alone can provide only 13% of the net December thermal requirement at a cost of $300,000 based upon $10/ft^2. The most efficient collectors can provide only 32% of the same December requirement at a cost of $677,000.

3.2.3 Total Energy Analysis

3.2.3.1 General

Instead of using the inexpensive flat plate collectors having a low ΔT, the advanced type having higher efficiency and higher

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TABLE 3-2
SURVEY OF FLAT PLATE SOLAR COLLECTORS

TABLE 3-2

MANUFACTURER / FACTOR	FALCO INC. (CA)	CALMAC (NJ)	E&K (WA)	CHAMBERLAINE (IL)	ACORN STRUCTURES "SUNWAVE" (MA)	COLUMBIA SOLAR (MA)	RAYPACK (CA)	HONEYWELL (IN)	SOLAR SYSTEMS "SOLARVAC" (TX)	GE "SOLAROUP"[1]	OWENS-ILLINOIS "SUNPACK"
COST ($/FT^2)	3.00	2.86	8.50	11.00 +	9 - 11.00	14.50	15.40	12.75	14.84	20.00	20.00
PROJECTED COST ($/FT^2)	?	?	?	?	?	?	?	?	—	'79:10.00 '80:5.00	82': 10.00
MAX FT^2/PANEL	40	500	48	21	80	22.6	21.4	16	32	17.4	32
HEAT GAIN (BTU/HR-FT^2)	24.3	27.8	—	SNGL 21 DBL 30.4	—		SNGL 16.5 DBL 30.4	SNGL 44 DBL 39	42	48.0	53.8
COST ($/BTU)	0.015	0.013	—	.065 .045	—		0.12 0.063	0.036 0.04	0.044	0.051	0.046
ABSORBER PLATE MATERIAL	CPVC	CU TUBES ON FIBERGLASS	CU TUBES	STEEL	CU TUBES	CU	CU TUBES ON Al	STEEL PLATE - CU TUBES	CU TUBES	CU TUBES	CU
SELECTIVE COATING	N.A.	$\delta = 0.94$ $\epsilon = 0.55$	NO INFO	$\delta = 0.9$ $\epsilon = 0.58$	FLAT BLACK	FLAT BLACK	$\delta = 0.96$ $\epsilon = 0.75$	$\delta = 0.94$ $\epsilon = 0.10$		$\delta = 0.85$ $\epsilon = 0.04$	$\delta = 0.86$ $\epsilon = 0.07$
NUMBER OF GLAZINGS	1	1	1 OR 2	1 OR 2	1	1 OR 2	1 OR 2	1 OR 2	1	2 (EVACUATED)	2 (EVACUATED)
MATERIAL OF GLAZING	?	KALWALL	CRYST GLASS	LOW IRON GLASS	ACRYLIC	PLASTIC	TEMPERED GLASS	LOW IRON GLASS	ACRYLIC	GLASS	GLASS
HEAT EXTRACTION MEDIUM	FLUID	FLUID	FLUID	CORROSION INHIBITED FLUID	WATER	FLUID - ANTIFREEZE	WATER	DOW THERM	ETHIGLYCOL	FLUID 30% ETHIGLYCOL	FLUID
INSULATION	NONE	NONE	NO INFO	FIBERGLASS	2" FIBERGLASS	2-1/2" FIBERGLASS	BACK & SIDES	3-1/2" BACK 1" SIDE	NOT NEEDED DUE TO VACUUM	FIBERGLASS VACUUM	1.5" POLYARETH
AVERAGE ΔT RISE (°F)	4	NOT SUPPLIED	6	—		10 (HI INSOL)	NO INFO	—		30	20 - 25
AVERAGE FLOW RATE (GPM)	4	3	2	0.33	2.0 - 2.5	0.6 - 1.0		0.3 - 0.7	0.85	0.22	0.3
LIFE EXPECTANCY (YR)	30	30	20	—			20	20	20+	15+	
FROM NBS CURVE X INTERCEPT Y INTERCEPT	 0.72 0.72	 0.77 0.80	NO INFO	SNGL 0.67 DBL >0.9 0.72 0.68			SNGL 0.6 DBL 0.84 0.78 0.76	SNGL 1.3 DBL 1.05 0.78 0.84	 1.4 0.72	 0.72 0.60	 >1.5 0.6
WARRANTY YEARS	10 (LTD)	5 (PRORATED)	2	1			5 (LTD)	NO INFO	1 (LTD)		1
APPLICATIONS	POOLS HOT WATER	POOLS HOT WATER WARM AIR	WATER	WATER HEAT AC	HOT WATER HEAT	HOT WATER	HOT WATER	HEAT HOT WATER	HEAT HOT WATER AC	HEAT AC	HEAT HOT WATER
SPECIAL FEATURES	DRAIN DOWN TOTALLY WETTED SURFACE	DESIGNED FOR EXPANSION & CONTRACTION LARGE PANELS LESS PLUMBING		90% WETTED SURFACE	DRAIN DOWN			NO FLOW TO 850°F	VACUUM SURROUNDING ABS PLATE. DRAIN DOWN SURVIVAL NO FLOW IN PA	SURVIVES NO FLOW TO 700°F. NON-IMAGING CONCENTRATOR BUILT IN	WITHSTANDS NO FLOW. RESISTANT TO THERMAL SHOCK. INDEPENDENT OF WIND
PROBLEMS OR CONSIDERATIONS	MAX T = 300°F NO FLOW	MAX T = 400°F NO FLOW			MAX T = 260°F		MAX T = 400°F NO FLOW				

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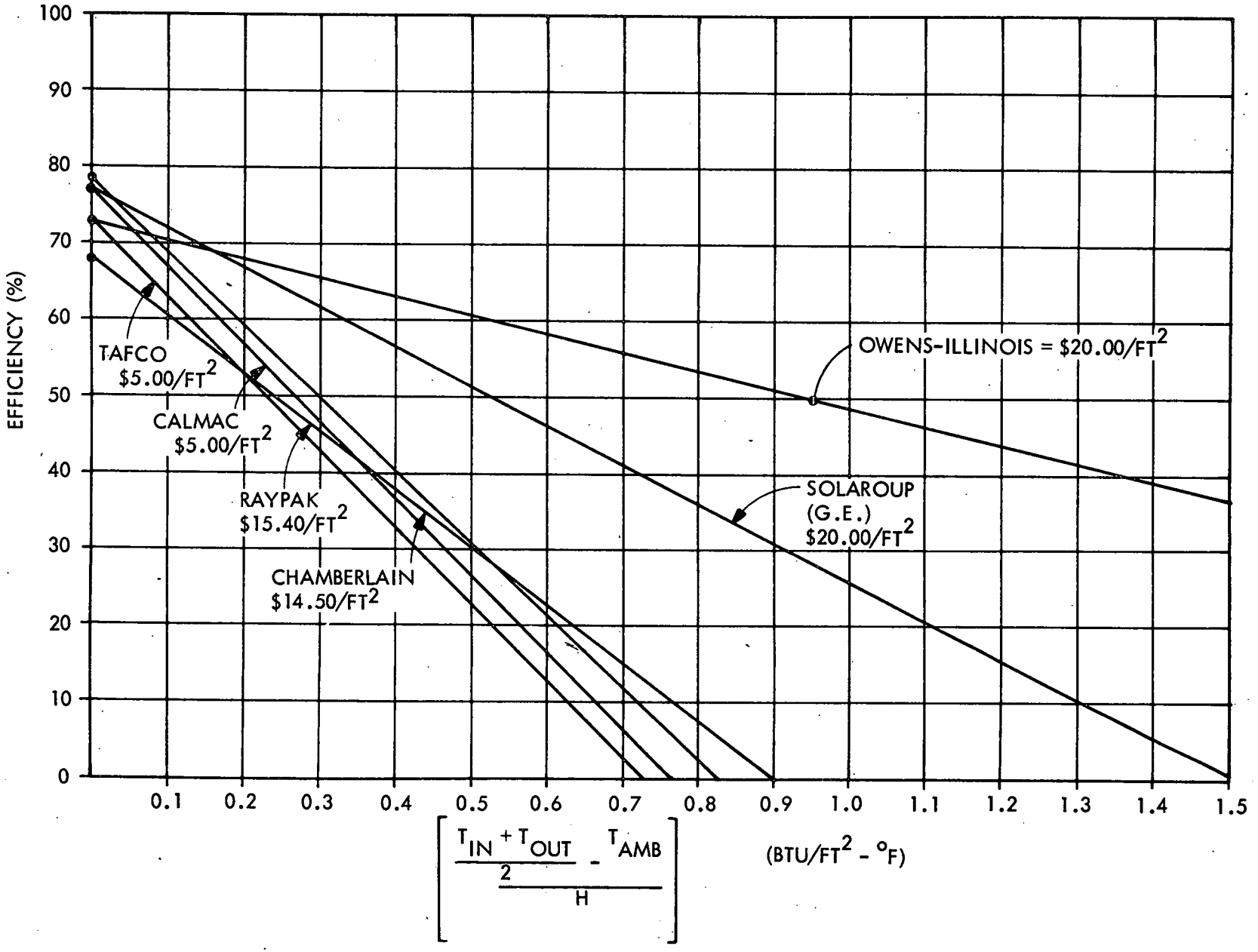


Figure 3-1. NBS Ratings for Flat Plate Solar Collectors.

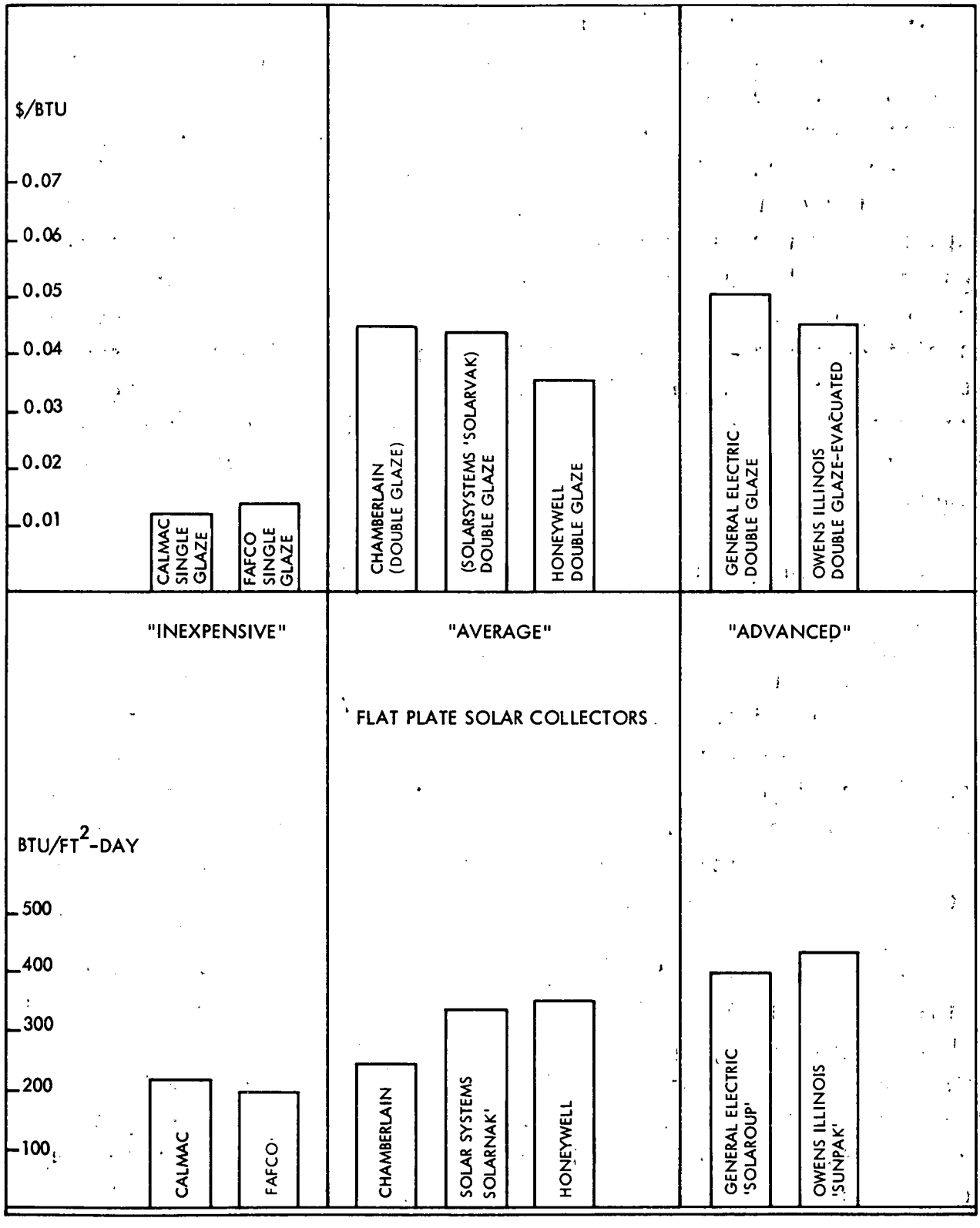


Figure 3-2. Comparison of Efficiency and Cost for Commercial Flat Plate Collectors.

temperature rise make the organic Rankine engine suitable for direct drive of pumps and/or generators. The thermal energy and the waste heat can then be used to meet the needs of the plant. In its simplest form, the system consists of a heat recovery boiler which transfers the heat from the solar collector to a Rankine cycle working fluid, providing saturated or super-heated vapor to the expander. Power is extracted in an expansion turbine as the fluid passes into the condenser of which provides liquid to the feedpump. The feedpump raises the pressure and supplies fluid to the boiler, completing the cycle (Figure 3-3). The working fluid condenser heat is rejected to the air. The expander shaft work can be transmitted through a speed-reducing gear box and ultimately used as shaft power or used to drive a generator to produce electrical power.

3.2.2.2 Fluid Selection

Considerable work has been done to select optimum working fluids for various applications. A large number of parameters influence the selection of the optimum working fluid. Flammability and toxicity for some applications are major considerations and greatly reduce the number of candidate fluids. In general, water is not a good working fluid for a low temperature Rankine cycle because its large latent heat of vaporization results in low boiling pressures and therefore low cycle efficiency.

Many fluids have been used or are being used in Rankine cycle systems. These fluids include Trifluoroethanol (Fluorinol), FC-75, MiPB, water, Refrigerant 113, Dowtherm A, monochlorobenzene, Pyridene and Toluene. The fluid selection, while an important detail of the final application, does not change the potential of a heat recovery system. However, limitations of a selected fluid may reduce the actual cycle efficiencies.

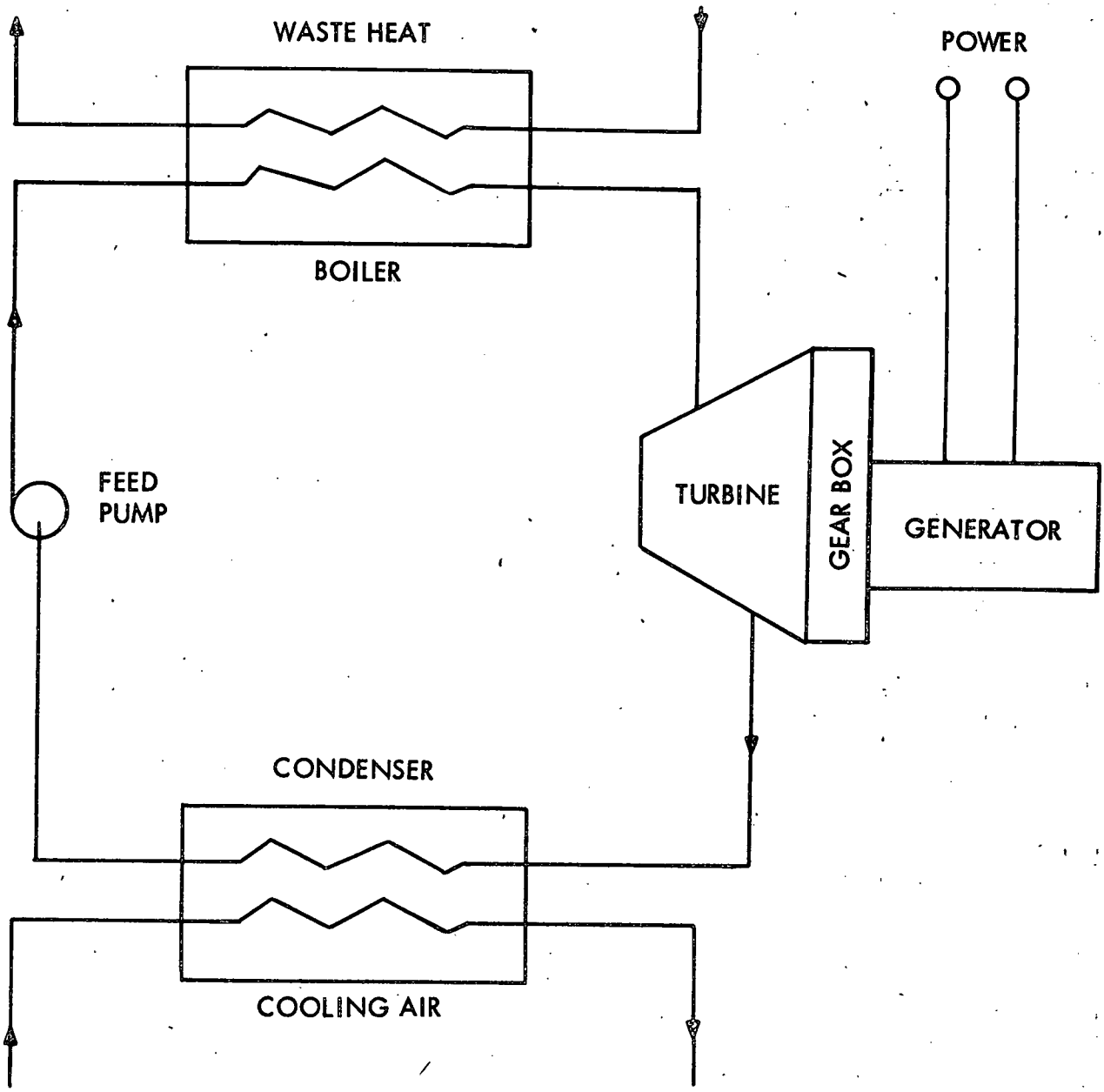


Figure 3-3. Heat Recovery System Schematic.

The organic fluid selected for this candidate Rankine cycle power plant is Freon-11. The characteristics of this fluid are shown in Table 3-3. The physical and thermodynamic characteristics of Freon-11 make it suitable for low pressure Rankine cycle systems.

3.2.2.3 Rankine Cycle Characteristics

3.2.2.3.1 Calculations

The Temperature-Entropy diagram of Freon-11 is shown in Figure 3-4. State points were set based on thermodynamic properties of Freon-11 in Table 3-3. As a result of the conditions set in Table 3-3, enthalpy balance was made to determine the available heat load, mass flow rate of Freon-11, cycle efficiency, turbine shaft power and cooling air requirements.

Table 3-4 defines symbols and subscripts used in these calculation. Taking control surface around each component and satisfying the first and second laws of thermodynamics, the mass flow rate and the power extracted from the bottoming cycle was determin d as follows:

$$q_{Exhaust} = \dot{m}\, c_p\, (T_o - T_i)_{Exhaust}$$

$$q_{Freon} = \dot{m}\, c_p\, (T_o - T_i)_{Freon}$$

Equating these two equations, the Freon-11 mass flow rate is solved for the prescribed temperature limits. For the control surface around the pump:

$$W_p = h_2 - h_1$$

$$S_2 = S_1$$

$$h_2 - h_1 = \int_1^2 v dP = v\,(P_2 - P_1) = W_p$$

TABLE 3-3

Freon-11 Characteristics

Physical Characteristics

Chemical Composition:	CCl_3F
Molecular Weight:	137.38
Freezing Point:	-168^oF
Boiling Point:	74.87^oF
Critical Temperature:	388.4^oF
Critical Pressure:	639.5 psia
Flammability	Fuming

Properties of Freon-11 at Various States of the Rankine Cycle

Condition	T (^oF)	P (psia)	H (Btu/lb)	S (Btu/lb^{o}F)	V_3 (ft/lb)
1	150	52	39.2	0.076	0.82
2	153	54	39.8	0.078	0.080
3	220	130	116	0.193	0.339
4	150	52	109	0.192	0.82

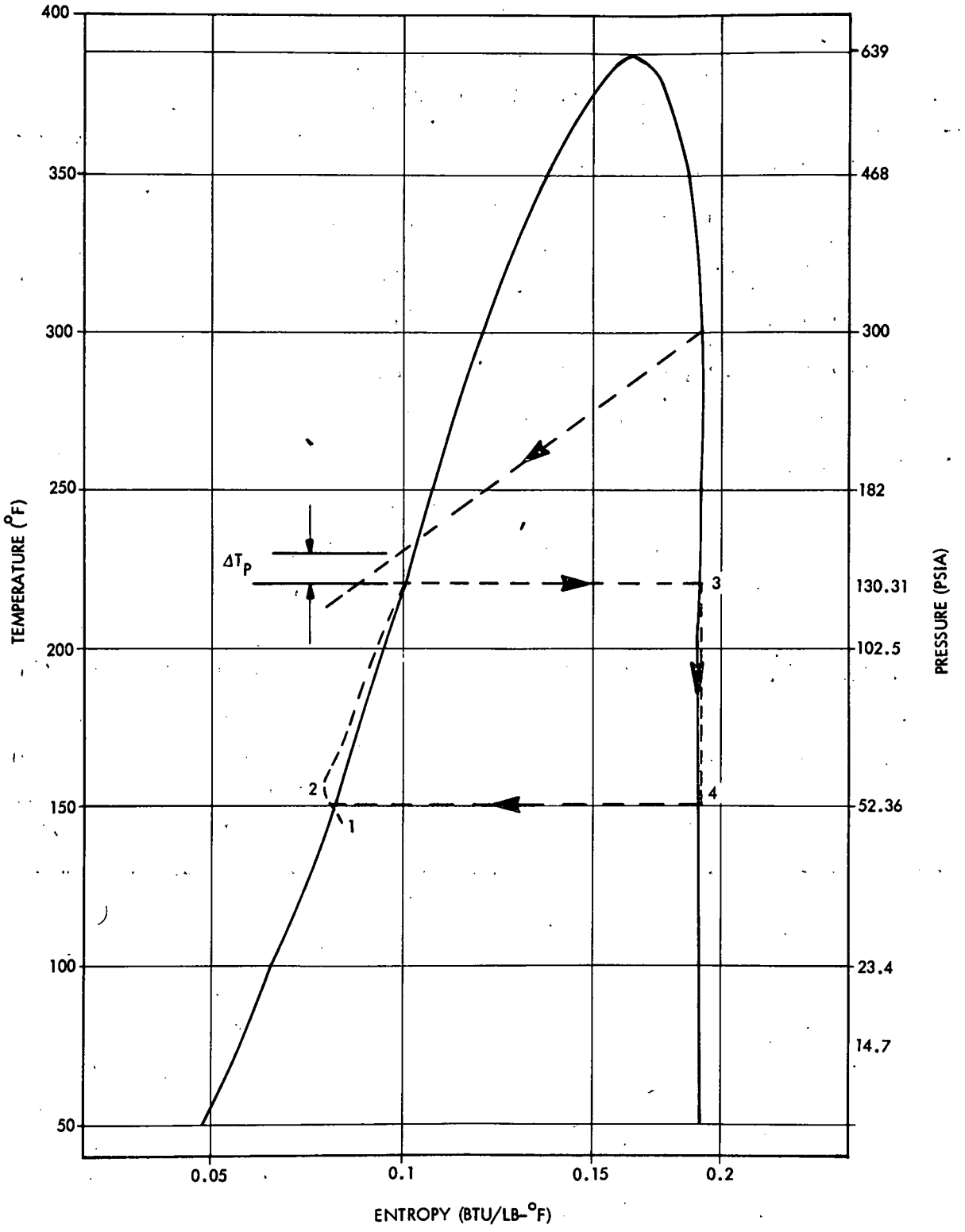


Figure 3-4. Temperature - Entropy Diagram of Freon-11.

TABLE 3-4

SYMBOLS AND SUBSCRIPTS

SYMBOLS

C_p:	specific heat at constant pressure Btu/lb^{o}F
h:	enthalpy Btu/lb
$\dot{m}$:	massflow rate lb/hr
q:	heat load Btu/hr
s:	entropy Btu/lb-^{o}F
t:	temperature (^{o}F)
v:	specific volume ft^3/lb
w:	work Btu/lb
x:	intermediate state between liquid and gas
n:	efficiency

SUBSCRIPTS

1,2,3,4:	state points on T-S diagram
i:	inlet
o:	outlet
f:	liquid phase
g:	gaseous phase
fg:	liquid-gas mixture
H:	heating
P:	pump
t:	turbine
th:	thermal
m:	mechanical
oV:	overall

From saturated liquid Freon-11 tables, h_1 and v_1 is determined as follows:

$$H_2 = h_1 + W_p$$

For the control surfaces around the boiler:

$$q_H = h_3 - h_2$$

From saturated tables of Freon-11 vapor, h_3 is determined.

For the control surface around the turbine:

$$W_t = h_3 - h_4$$

$$S_3 = S_4$$

The quality of vapor at State (4) is now determined:

$$S_3 = \left[S_g - (1-x)\ S_{fg}\right]_4$$

where

$$S_{fg} = S_g - S_f$$

The entropy values at States 3 and 4 are determined from the Mollier diagram of Freon-11, hence the vapor quality(x) is solved. Similarly,

$$h_4 = \left[h_g - (1-x)_4\ h_{fg}\right]_4$$

Substituting the vapor quality and enthalpy values of h_f and h_{fg} at State 4, the corrected enthalpy is determined. The turbine work is given by:

$$w_t = h_3 - h_4$$

and net work

$$w_{net} = w_t - w_p$$

$$n_{th} = \frac{w_{net}}{q_H}$$

At the approximate optimum operating cycle conditions, some variations on the boiling and condensing conditions were introduced to determine their effect on cycle efficiency.

3.2.2.3.2 Peak Cycle Pressure Effect

Figure 3-5 shows the effect of expander inlet pressure on cycle efficiency for conditions where superheating is not present. The results are exact for the conditions shown on the figure and are qualitatively correct for any expander efficiency, pump efficiency or condensing conditions. It can be seen from the figure that for a fixed inlet temperature, as cycle pressure is increased cycle efficiency will also increase and then start to decrease. This decrease is caused by the additional pump work required to obtain the higher pressures, more than off-setting the increased head available to the expander. The figure shows that 130 psia at an inlet temperature of 220°F is optimum for use in cycle efficiency calculations, and that near this inlet pressure, the exact pressure selected has only slight effect on cycle efficiency.

3.2.2.3.3 Condensing Temperature Effect

The choice of condensing temperature influences cycle efficiency and heat exchanger volume. From Figure 3-6 which is a plot of cycle efficiency versus condensing temperature, cycle efficiency increases as condensing temperature decreases because of the increased heat available across the expander. In addition, as the condensing temperature is decreased, the fan work increases because of the increased air flow required to achieve adequate heat transfer in the condenser. These effects cause a peak in the cycle efficiency as shown in the figure. As the condensing temperature is reduced below approximately 120°F, the fan work becomes a very significant percentage of the total expander output, reducing overall cycle efficiency.

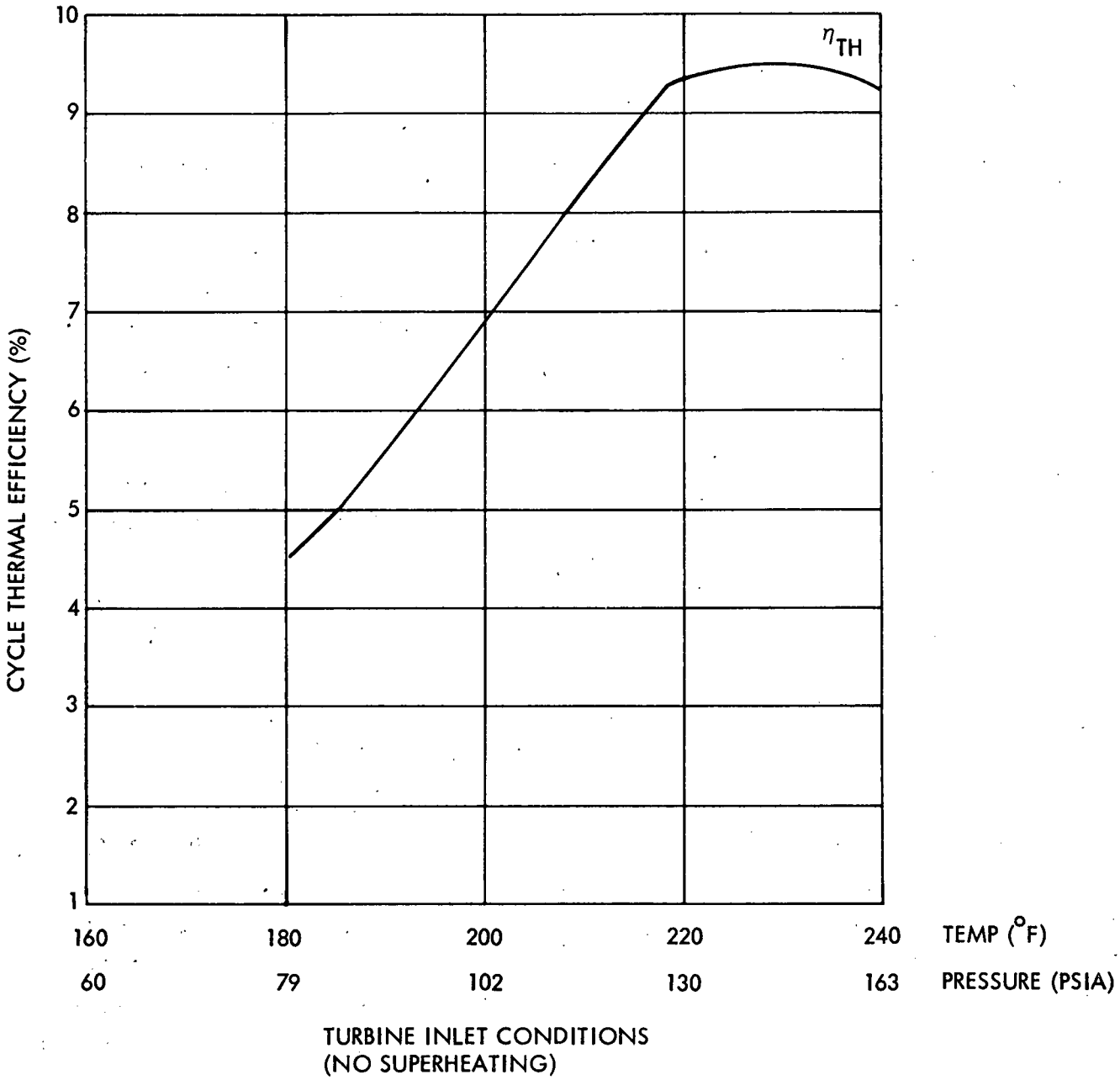


Figure 3-5. Expander Inlet Pressure Effect on Cycle Efficiency.

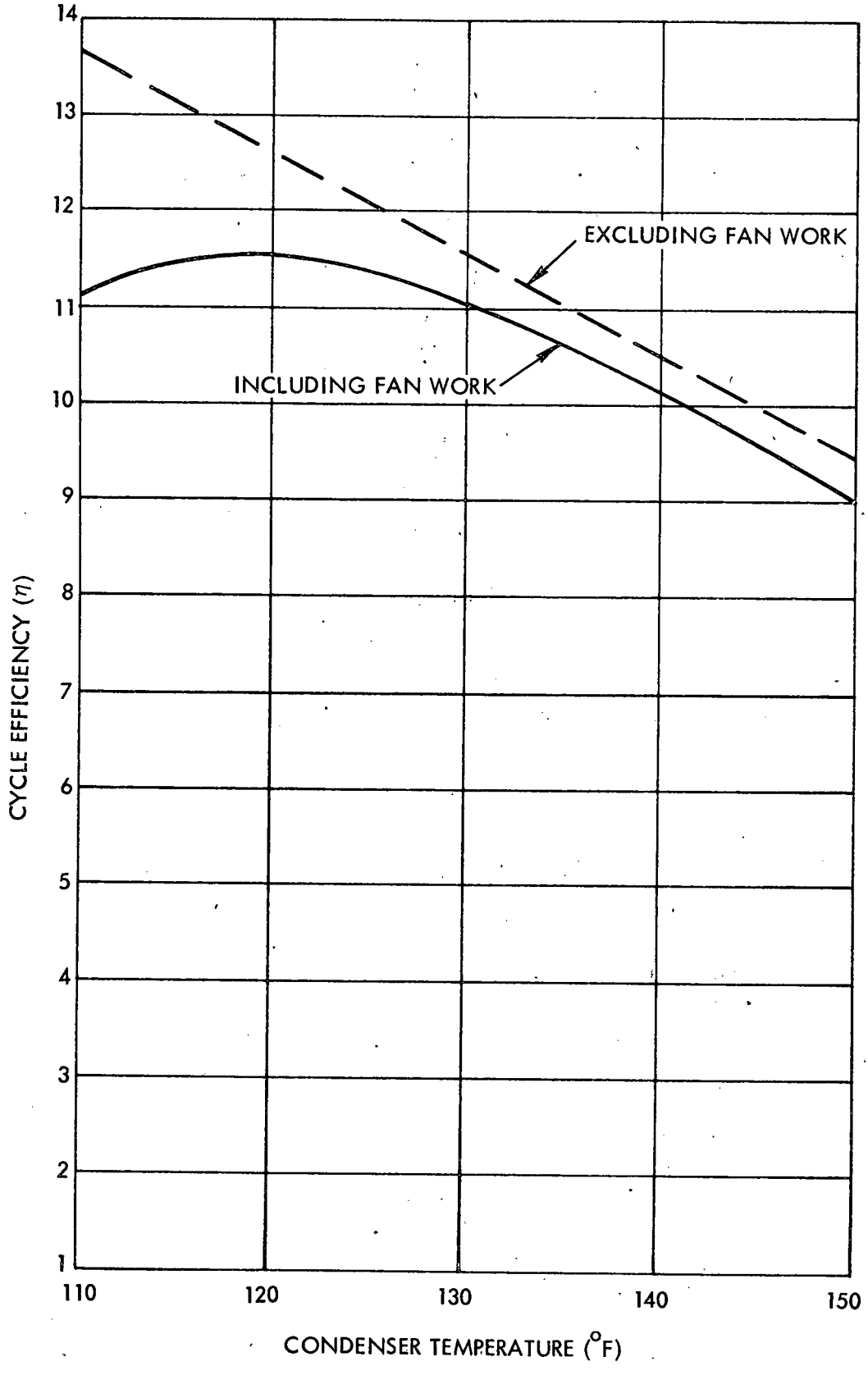


Figure 3-6. Condenser Temperature Effect on Cycle Efficiency.

With increased condensing temperature, the condensing volume decreases and boiler volume increases. This occurs because of the increased flow rate caused by the reduction in cycle efficiency. It has been found that with air-cooled condensers the minimum heat exchanger size occurs with condensing temperatures between 140 - 160°F and with water cooled condensers between 100 - 120°F. The condensing temperature at which the minimum heat exchanger volume occurs is not only a function of the cycle conditions selected, but also of the boiler hot gas inlet temperature and the temperature of the cooling water.

3.2.2.3.4 Expander Inlet Temperature Effects

For a fixed power and working fluid, increasing the expander inlet temperature increases the cycle efficiency. The maximum fluid temperature and pressure can be limited by the hot gas temperature and boiler design.

In the case under consideration where there is no superheating of the organic fluid, the effect of temperature (pressure) increase on cycle efficiency is clear from Figure 3-5.

3.2.2.4 Design Specifications

Based on the cycle parametric calculations, the design specifications for a power plant are shown in Table 3-5. The peak cycle temperature was chosen to be 220°F (consistent with advanced flat plate collectors) with a condensation temperature of 150°F. At this level, the bottoming plant would have a thermal efficiency of 9.5% and generate 344 kw. Because of the low efficiency, such a plant will require a mirror field capable of collecting nearly 5000 kw of solar heat which is far more than that required to meet both the electrical or thermal needs. Therefore, it is obvious that efficiency must be improved.

TABLE 3-5
EXHAUST HEATED RANKINE CYCLE SYSTEM
CYCLE CHARACTERISTICS

WORKING FLUID:	Freon-11
Mass Flow Rate:	53 lb/hr
BOILER:	
outlet pressure	130 psia
outlet temperature:	$220^{o}F$
Q_H (Total):	14.7×10^6 Btu/hr
Q_H (Sensible):	2.9×10^6 Btu/hr
Q_H (Latent):	11.8×10^6 Btu/hr
TURBINE:	
η_{th}:	0.85
η_m:	0.95
η_{oa}:	0.808
W_{sh}:	7.316 Btu/lb
Exhaust Temperature:	$150^{o}F$
Exhaust Pressure:	52 psia
CONDENSER:	
Pressure:	52 psia
Temperature:	$150^{o}F$
Q_c	13.3×10^6 Btu/hr
FEEDPUMP:	
η_{ov}:	0.70
Fan Power:	86 kw
Net Power Out:	258 kw
Cycle Efficiency:	9- 9.5%

One way of increasing efficiency is to operate the plant at higher temperatures. However, to reach higher temperatures, concentrating solar collectors are required. Simple concentrators can reach temperatures of 600°F where the organic Rankine cycle could be used with efficiency improvements from 9.5% to 25%. Above 600°F, the organic fluids decompose and super-heated steam becomes the only choice in Rankine power plants. Efficiency is low for steam plants and can reach 25% only at the upper temperature limit of 1200°F. Above 1200°F, the Brayton cycle becomes the most efficient type of power plant. At 2000°F a cycle efficiency of nearly 50% can be obtained.

3.2.4 Summary

A total energy plant has the potential to generate the 343 kw with high efficiency and to use most of the waste heat to supply the thermal demands of seawater and building during the winter months. The total system concept must now be cost analyzed and compared with flat plate collectors.

3.3 CONCENTRATING COLLECTORS

3.3.1 General

A cost analysis was undertaken for total energy system which will have maximum efficiency; i.e., a Brayton cycle power plant operating at 2000°F incorporating a recuperator and energy storage. With a fossil-fuel option, the amount of solar energy; i.e., solar collection area and storage, can be optimized on the basis of operating cost and capital investment.

In small sizes (up to 50 Kwe), high concentration ratios can be economically achieved with a single tracking telescope. However, for larger sizes, a mirror field is needed to concentrate 1500 -2000

solar images on a receiver mounted on the top of a tower. Within the tower is the storage stove and power generator equipment. The tower and mirror field should be located to the north of the building in an area clear of shadows from tall trees and buildings.

3.3.2 Brayton Cycle

The gas turbine engine used for aircraft propulsion has accounted for most of the Brayton cycle engine developmental effort, which started in the 1930's. The aircraft application emphasized a high thrust, lightweight design rather than high efficiency. A low pressure ratio recuperated cycle having a higher weight to power ratio improves efficiency over the aircraft type and becomes a primary candidate for the ground-based power plants. There are few models of the recuperated cycle engines used experimentally for gas turbine-driven trucks and buses. A comparison of performance characteristics of both types of gas turbines (Figures 3-7 and 3-8) at temperature ratio of 5 shows the recuperated cycle to have a peak efficiency of 52% at a pressure ratio of only 4, whereas, the non-recuperated cycle has its peak efficiency of 43% at a pressure ratio of 32.

A power plant designed to operate under these conditions is available only on an experimental basis. The desired gas turbine performance specifications (Table 3-6) is obtainable with present technology and represents design goals for a total energy system.

TABLE 3-6 PERFORMANCE SPECIFICATIONS

Pressure Ratios:	4:1
Temperature Ratio:	4.7:1
Compressor Efficiency:	90%
Turbine Efficiency:	90%
Recuperator Effectiveness:	90%
Sum of Total Pressure losses $\frac{\Delta p}{p}$:	6%
Airflow Leakage:	5%
Cycle Efficiency:	48.3%

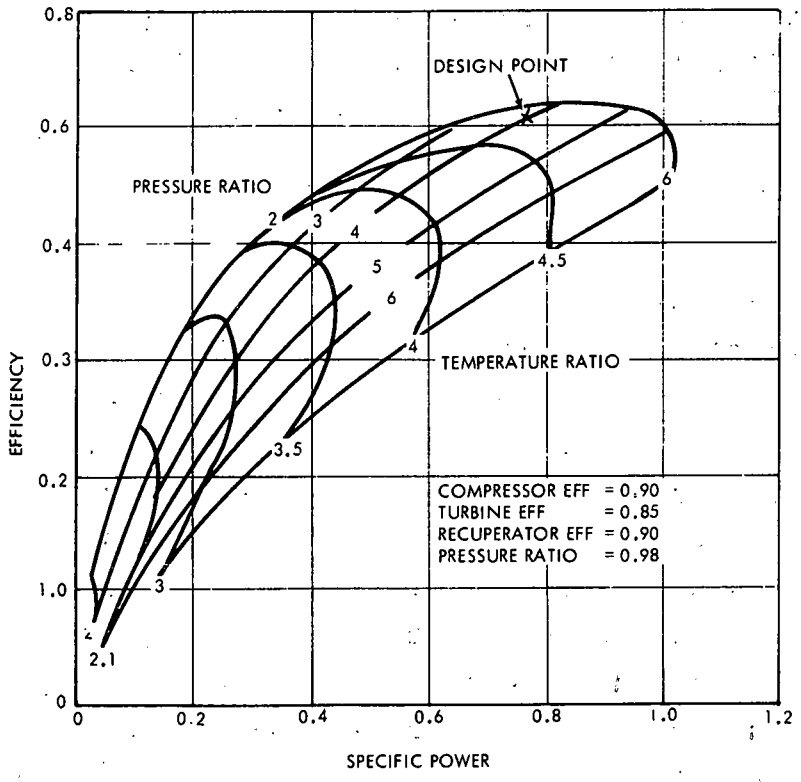


Figure 3-7. Theoretical Performance of Recuperative Gas Turbine Engine.

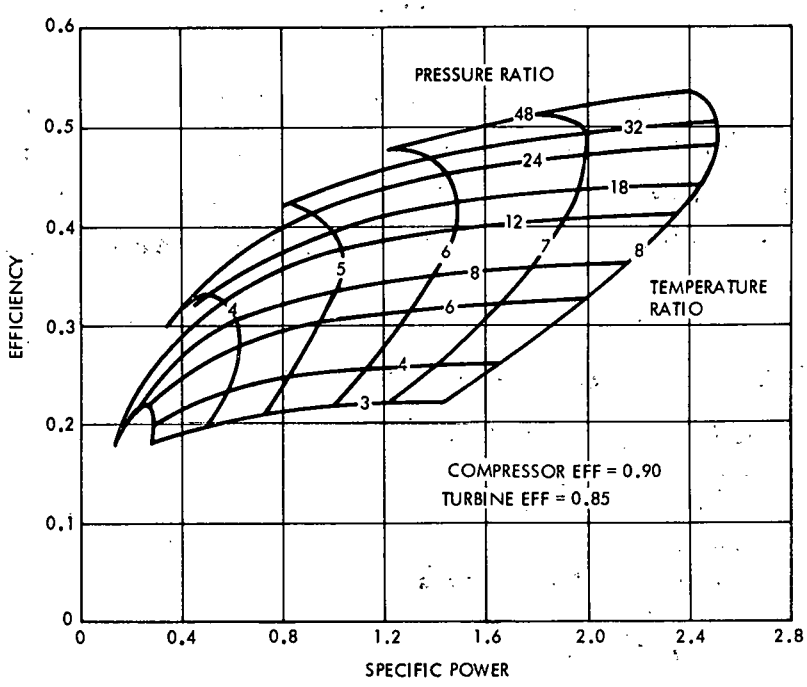


Figure 3-8. Theoretical Performance of Aircraft-Type Gas Turbine Engine.

The choice of recuperators varies from the large metal fabrications using multiple, small adjacent tubes which produce efficient counterflow heat exchange between gases, to rotating ceramic wheels which are alternately heated and cooled as the wheel rotates across channels of hot and cold gas. Rotating wheels have been fabricated by Ford Motor Company for automobiles and by Corning for gas turbine trucks and bus engines. Garrett, General Motors, Williams and Chrysler all make engines at the approximate 400 horsepower level, well within the aquaculture power plant requirement.

The cost of a Brayton engine power plant, including generator and controls, in this size will be less than one half of the cost of the Organic Rankine plant, or $120 per kilowatt-hour. Cost savings accrue from the open cycle, one less heat exchanger and a great reduction in plumbing. Not included in this cost estimate is the possibility of large production Brayton truck and bus engines which will further reduce the power plant cost.

3.3.3 Mirror Field

The mirror field design, Figure 3-9, is elliptical on the earth's plane with the tower located south of field center. The outward lines, like spokes of a wheel, give the bearing or azimuth angle. The concentric circles about the tower represent the elevation angles between the heliostats and line-of-sight of the receiver. The 60 degree pie segment located directly behind the tower is blocked by the cylindrical concrete tower and does not contain mirrors. All mirrors are individually controlled to focus the sun at the cavity in the receiver. Around the receiver, whose axis is tilted 14 degrees to the north, the mirrors are grouped, separated from each other so as to minimize blocking during the day, and concentrated within a half angle of 62 degrees about the cavity.

The mounting and motions of the mirror system are patterned after those being used at the Georgia Institute of Technology in that solar test facility. Previous mirror studies indicate a packing density of the mirrors of approximately 39%.

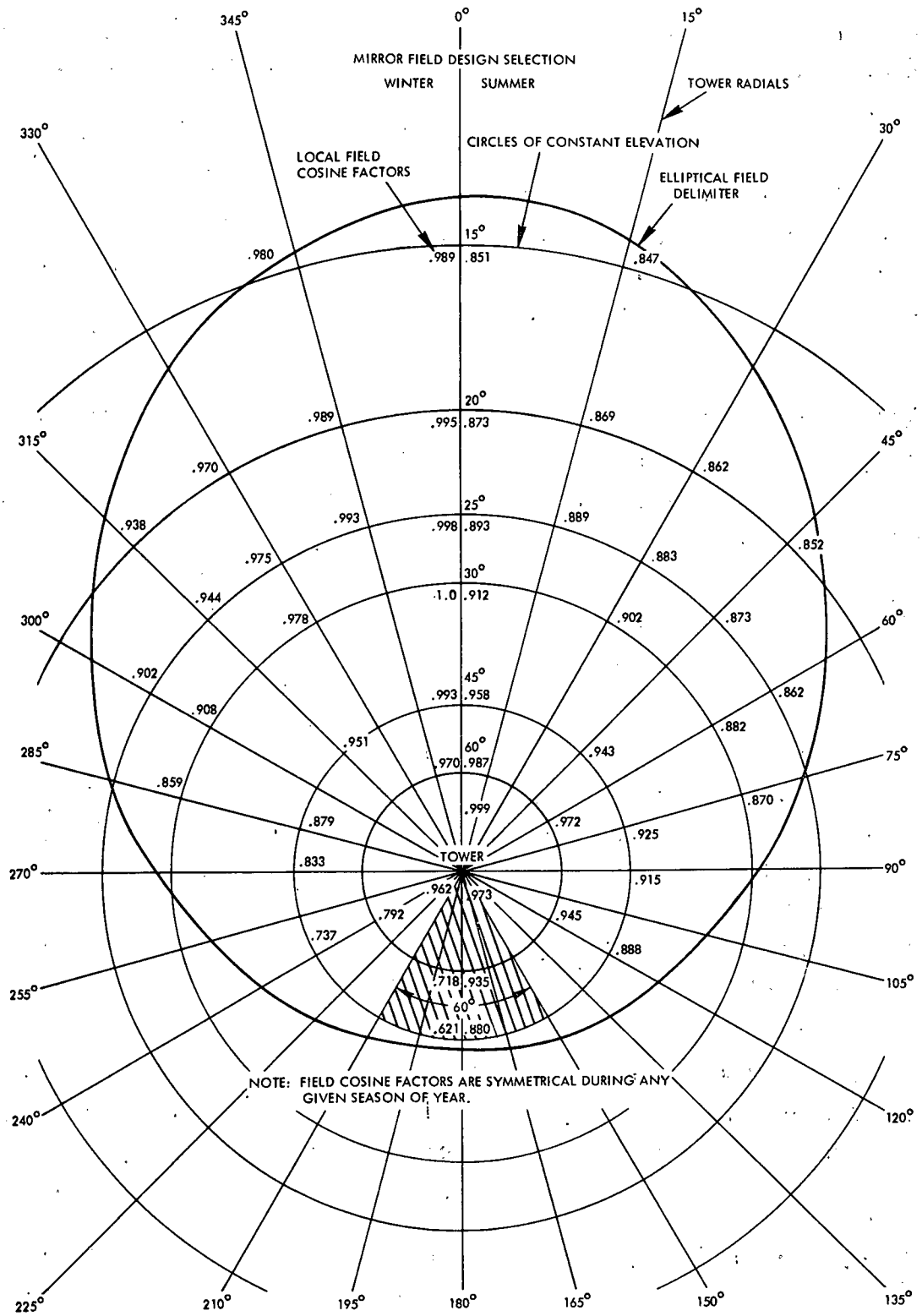


Figure 3-9. Mirror Field Design.

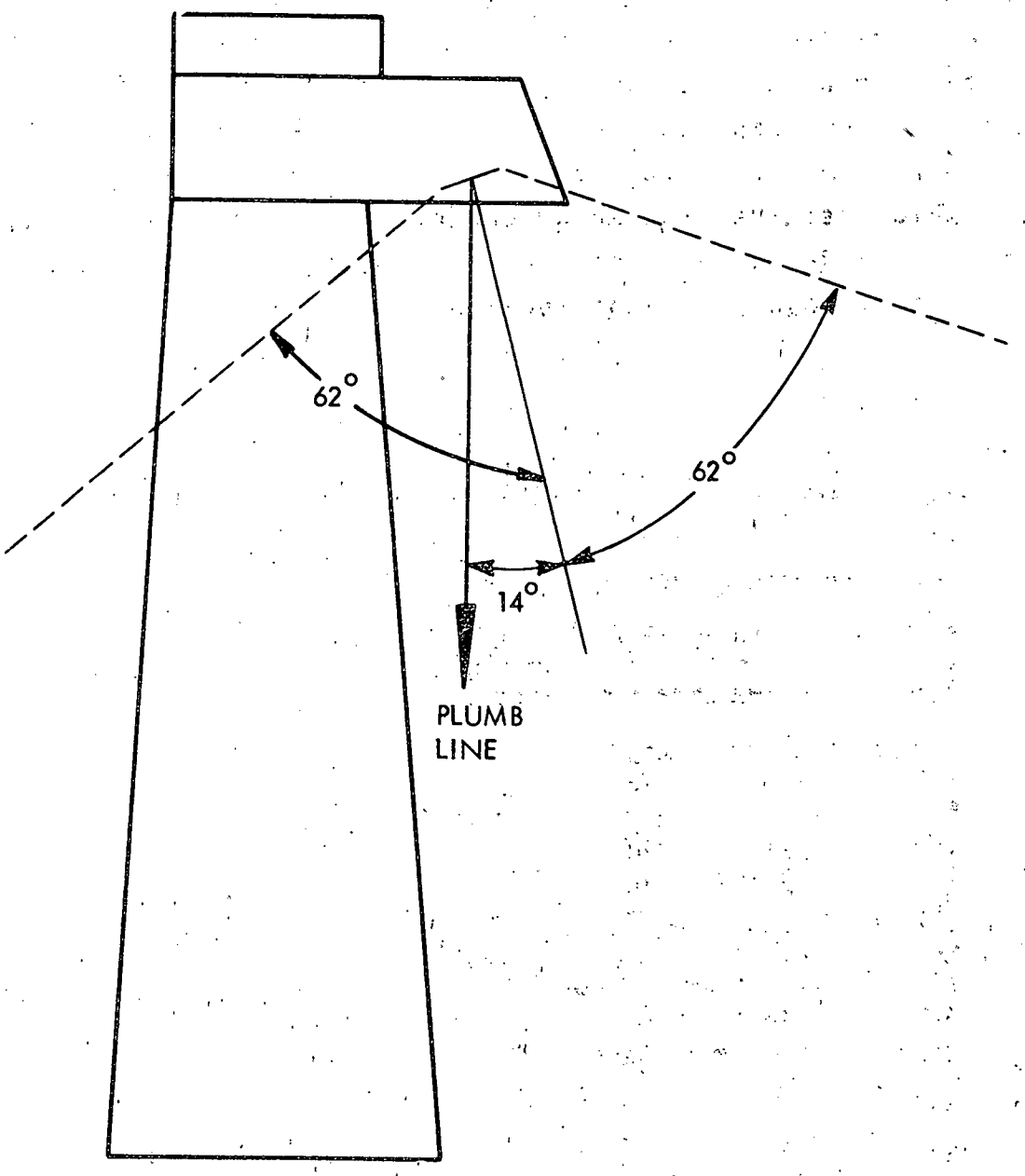


Figure 3-10. Tower Design

3.3.4 Receiver/Tower Design

3.3.4.1 General Arrangement

The tower positions the receiver over the mirror field with a clear field-of-view of the cavity from the ground except from the 60 degree quadrant to the south of the tower (Figures 3-9 and 3-10). The active system components are located in the enclosure at the top of the tower. On the north side of the enclosure is a cavity receiver which receives the solar energy from the mirror field, as shown in Figure 3-11. Storage stoves are located at the same level and are closely coupled to the receiver.

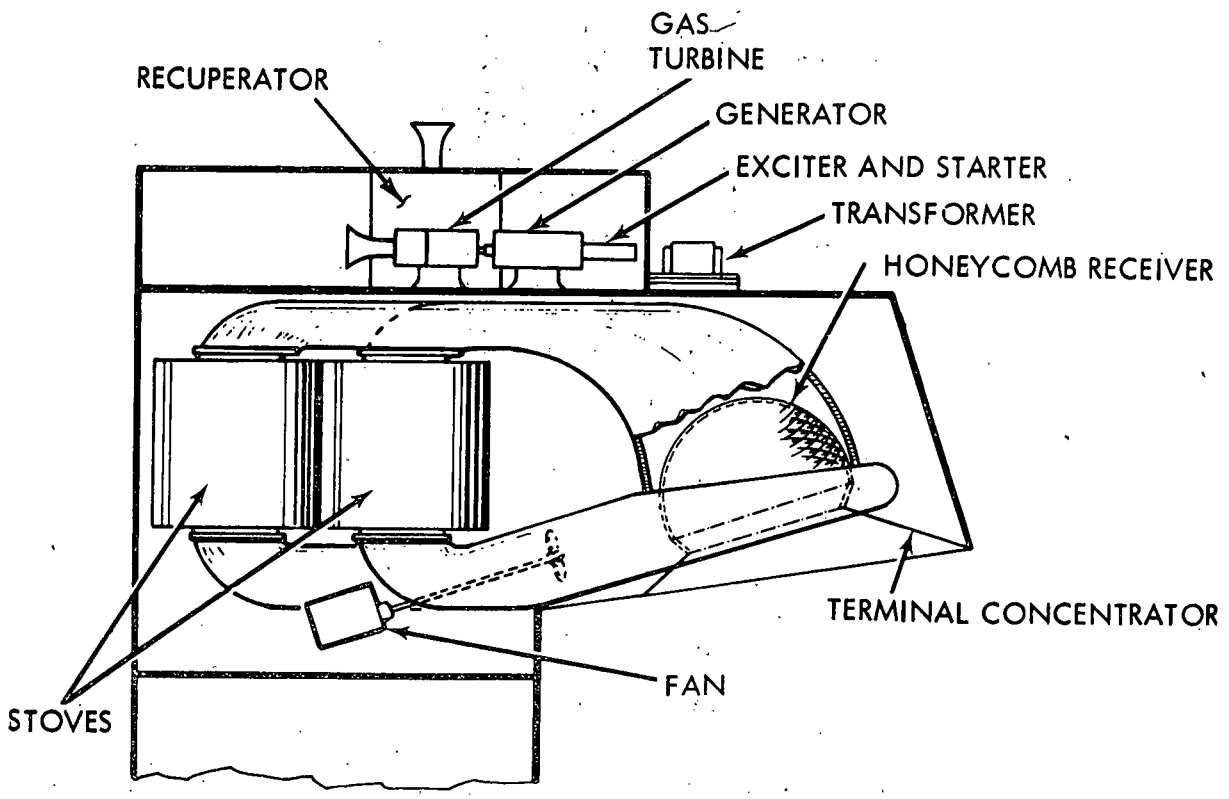


Figure 3-11. Power Storage Tower

Receiver and cavity designs are based on heat transfer analysis and experimental results obtained from a 10 KWth unit tested by Sanders in the solar test facilities at White Sands Missile Range. Within the cavity is a honeycomb solar collector made from bricks of silicon carbide (SiC) assembled into a geodesic dome. The manifold surrounding the base of the receiver supplies a controlled amount of air to the cavity. As air passes through the honeycomb, it is heated and ducted to the top of the checker stove. A variable speed fan at the bottom of the checker stove controls the airflow such that the air temperature entering the top of the checker stove is maintained at 2000°F. A wind shield extending beyond the cavity entrance protects the cavity against wind convection losses and also functions as a terminal concentrator to deflect part of the reflected beam from the farthest mirrors into the cavity.

Sanders has examined the cost of its solar receiver which includes the circulator fan, ducting, honeycomb, insulation and terminal concentrator. Over a wide range of sizes, the cost is approximately $42/KWth. In this case, the thermal specifications are derived from the size of the mirror field and not the production plant power.

The gas turbine is positioned directly above the stoves. This location has the advantage of clean, cool inlet air for the most efficient gas turbine cycle, plus short routing to minimize pumping and convection losses. In addition, the inlet air, which would require filtering if the inlets were located at ground level, is free of sand and dirt.

The heavy equipment such as stoves, electric motors and ducts are positioned directly above the cylindrical tower support. The lightweight receiver is cantilevered to the north.

Air for the Brayton machinery enters the gas turbine at ambient conditions (Figures 3-12), is compressed to four atmospheres, and

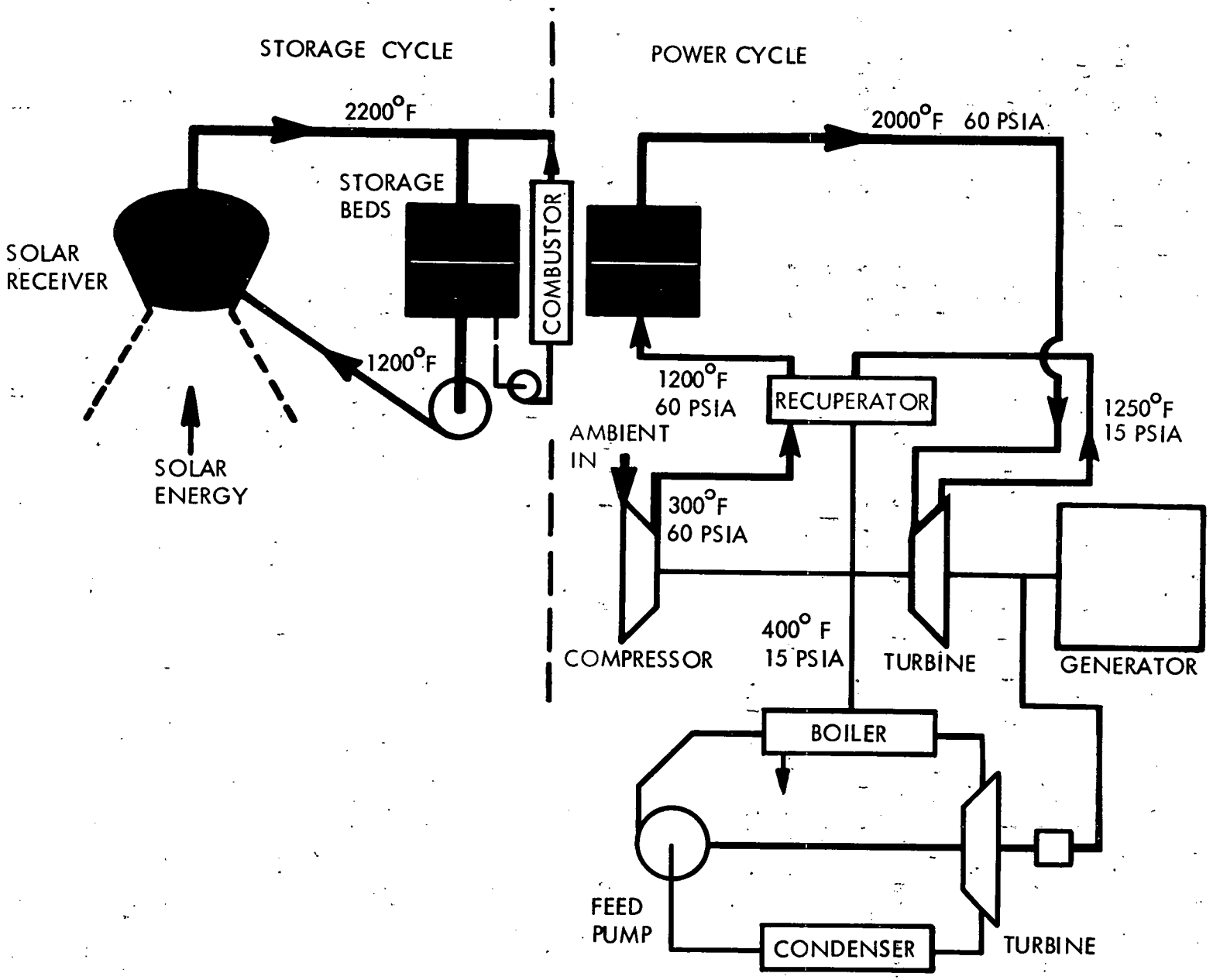


Figure 3-12. Brayton - Air Cycle Block Diagram

is directed down to the bottom of the fully charged stove. As the air rises through the stove, it absorbs heat from the firebrick and returns to the gas turbine. After expansion through the turbine, the hot gases are at near-ambient pressure as they enter the recuperator. The air leaving the recuperator then enters a waste heat boiler to generate heat to run the Rankine cycle turbine connected to the generator. A condenser and feedpump complete the block diagram.

The salient feature of this general arrangement is that the solar collection cycle functions independently of the power generation cycle. In this way, the energy collection can be performed at ambient pressure and the power generation at the optimum operating pressure for maximum cycle efficiency.

3.3.5 Storage

3.3.5.1 Checker Stove Description

The checker stove is a heat exchanger device used in the glass and steel industries since the early 1800's. As shown in Figure 3-13, it consists of an insulated pressure shell approximately 11 meters (36 ft) in diameter and 40 meters (130 ft) high, containing internal air ducts and a large array of refractory bricks called checkers. The checkers are arranged in stacks, often 30 meters (100 ft) high, forming a large number of individual air passages called flues, through which air can flow. Heat is alternately stored in the checkers, or removed from them during opposing portions of the process cycle.

In the steel industry these stoves are used to supply vast quantities of very hot air into the blast furnace charged with iron ore, coke and limestone. Heat and carbon monoxide released from the coke reduce the iron ore, while the limestone absorbs

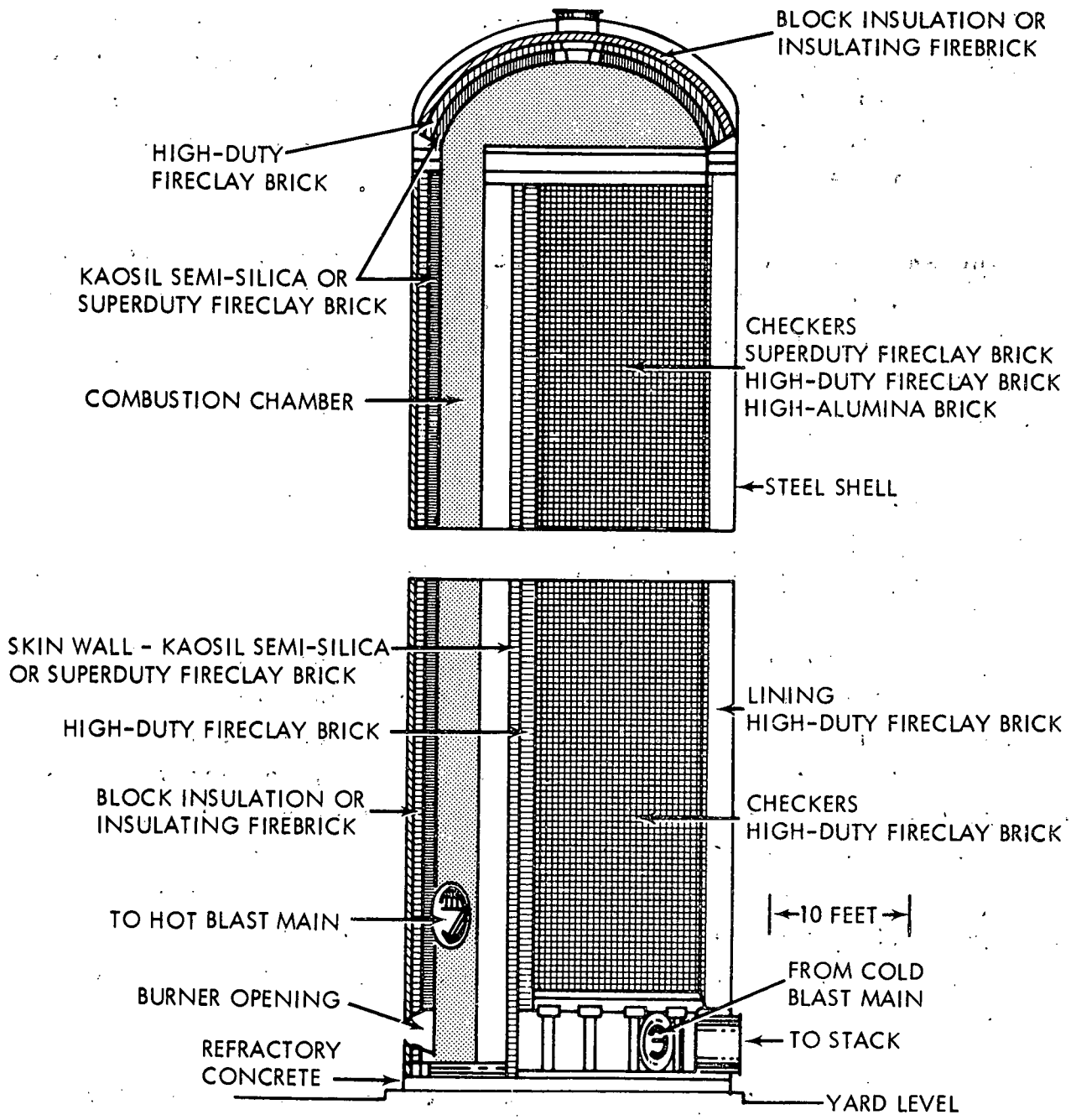


Figure 3-13. Typical Blast Furnace Stove.

various impurities. The hot flux of the furnace is piped into another "cold" stove where heat is extracted for use during the next blast period. At a typical installation, a furnace would have three or four stoves manifolded together with automatic valves, with one or two stoves "on blast" while the others are "on gas" or "on charge". Through an arrangement called staggered parallel operation, the stoves are valved from one position to another to maintain constant output temperatures during the continuous operation of the furnace.

To maximize manufacturing efficiency by reducing the amount of coke required, the blast temperature has been progressively increased. Currently an output temperature of at least 2200°F is generated, which is also ideal for efficient gas turbine operation. Furthermore, typical stoves deliver this air at 3 to 4 atmospheric pressure, as required by the furnace combustor. This pressure is nearly that required if the stove were to exhaust into a turbine designed for recuperated operation. Operating a gas turbine from a pressurized checker stove was first reported in England. This configuration has the unique capability of handling large flow rates with low internal pressure losses, a factor which minimizes pumping losses and maintains high Brayton cycle efficiency.

3.3.5.2 Efficiency and Cost

The primary losses in efficiency are: (a) the thermal losses through the piping and container walls, and (b) the pumping losses incurred in charging and discharging the stoves. Pumping losses are difficult to estimate at low fluid velocity, but a conservative assumption was made that losses equalling two times the velocity head were possible; the maximum velocity is that passing through the large valves. Allocating the same losses to both the charging and discharging cycle, the pumping losses total to nearly 2%. A total loss of 5% for the storage function covers both the thermal losses, estimated at less than 1% using two feet of Babcock and Wilcox insulating firebrick (K-23), and the pumping and transient losses.

Cost estimates were obtained from Harbeson-Walker, a major manufacturer of checker stoves. A commercial firebrick and steel vessel insulated with firebrick was costed at $11.06/kwh$_{th}$. The three stoves representing 24,000 kwh$_{th}$ are also part of the concept.

3.4 PLANT COST ANALYSIS

3.4.1 Performance Specification

In estimating the performance expected from this system design, efficiency factors must be developed for all the operating steps in the power cycle (see Table 3-7). Although there is no engine available commercially which can perform to these specifications, there are many instances where comparable component efficiencies have been achieved and there remains only incentive to obtain these results. A cycle efficiency of 48% which, when combined with the mechanical-to-electric efficiency, yields an overall efficiency of 44%, a realistic goal for a Brayton cycle engine. Results from receiver tests at White Sands Solar Test Facility have demonstrated the feasibility of an 84% overall receiver efficiency.

There are two bottom line efficiencies depending upon whether or not there is an organic Rankine cycle engine operating from the exhaust of the Brayton engine. The total power plant cost is substantially increased by adding the Rankine engine, but this cost distributed over many years is more than compensated for by the reduction in mirror cost. A total power plant cost was estimated at $50,400 with the individual power plants costing approximately the same, although 2/3 of the power came from the Brayton engine. The unit costs used in estimating the power plant cost are as follows:

Two Rankine engines of 120 kw each @ $60/kw:	$14,000.
Two Brayton engines of 240 kw each @ $30/kw:	14,400.
One generator and controls for 360 kw @ $60/kw:	21,600
Total	$50,400

Note that one engine is on standby at all times.

TABLE 3-7

EFFICIENCY SCHEDULE

	Optimum Design
Engine Characteristics	
Pressure Ratio:	4:1
Turbine Inlet Temperature:	2000°F
Temperature Ratio:	4.7:1
Cycle Efficiency:	0.48
Turbo-Alternator:	0.44
Receiver	
Concentration Ratio:	2000
Maximum Temperature:	2200°F
Radiative Efficiency:	0.90
Insulation and Pumping:	0.96
Aperture Spillage:	0.98
Receiver:	0.84
Storage:	0.95
Tracking, Aiming, Shading, Blocking:	0.86
Reflection:	0.91
Overall Efficiency:	0.275
Overall Efficiency by adding a Bottoming Organic Rankine Cycle Power Plant	0.316

The mirror field is offset to the north patterned after studies made by Sandia Corporation. Efficiencies and packing densities were developed in the Sandia studies and are used here. The mirror system consists of individually controlled mirrors, each with its own calibration and tracking capability. Cost is estimated at $\$15/ft^2$ which includes a $\$3/ft^2$ assigned to the tower structure.

3.4.2 Economic Assumptions

Few assumptions have been made in projecting the total energy cost of the commercial sized plant. These boundary condition assumptions define the economic environment. Energy costs derived as such are meaningful and may be compared to alternate energy sources. The following paragraphs outline several economic assumptions and their resulting impacts.

Assumption: Residual plant value at the end of lifetime is zero; i.e., it is fully depreciated.

Impact: This maximizes capital amortization costs and increases apparent energy cost. The depreciation allowances which would be realized under present tax laws could effectively reduce capital amortization by up to 48%, the maximum corporate rate for earnings over $25,000. In this particular (aquaculture) application, these savings might be realizable even before commercial sales commence because new IRS regulations require periodic inventory of the bio-stock which then appear as earnings (asset growth).

Assumption: The investment tax credits are ignored.

Impact: Costs calculated herein are up to 10% higher for the portion related to manufacturing equipment and energy. Building, property, and plant equipment (plumbing, sanitation, space heating related) are not eligible for this requirement. That portion of

the plant generating electrical energy, and process heat are eligible. That grey area of waste heat utilization for building space heating is no doubt contestable and related equipment may or may not be considered manufacturing equipment.

Assumption: The secondary investment tax credits for energy conservation equipment - a part of the Energy Bill presently before the U.S. Congress is ignored.

Assumption: Property taxes range from $2.20/M to $156.25/M of real valuation depending on location within Maine (1976). The rates will more likely increase from the $21.00/M average 1976 as local budgets expand to meet the cost of expanding services and population. Taxation rates will probably increase at a rate greater than the economic inflation rate.

Impact: This expense, which may be considered certain, will increase the de-inflated yearly tax cost, at a rate probably between the economic inflation rate and the fossil fuel inflation rate.

Assumption: Excess collectible energy during Spring, Summer and Fall is neither sold nor stored in seasonal quantities, though energy is stored in daily quantities.

Impact: The prorated energy costs do not allow for short term perturbations (for example, 4 days of rain in May) but do reflect monthly averages. Ability to integrate short term deficiencies and, thus, save fuel required increased storage capacity. In either case a cost is incurred.

The optimum storage volume will satisfy daily storage requirements plus a fraction of the additional storage requirements caused by weather perturbations, which may be statistically modeled. Final determination of the optimized storage volume which will minimize combined storage, mirror field, and supplemental fuel costs

depends on a system sensitivity analysis which would be performed prior to implementing the system. The storage volume selected and priced for this system is on the size of the mirror field in each example and reflects the present cost of Number 6 fuel oil, 30.6¢/gal, and a projected annual fuel inflation rate of 10%. Fuel cost inflation at a greater rate would improve the economics of storage.

The net result of these assumptions is a conservative analysis which, nevertheless, shows the solar total energy system to be cost effective. Tax benefits could reduce energy costs by 50-60%. Property tax costs can be minimized by locating the plant within jurisdictions having and projecting low tax rates.

Inflation rates for fuel, the general economy and taxes are not assumed, but are parametrized. It is noteworthy that the absolute interest and inflation rates are not particularly important, but rather their relative values determine the effective energy costs.

3.4.3 Methodology

The energy cost projections result from a methodology wherein:

- Approximate energy requirements are determined.
- Energy system is designed and costed.
- Energy requirements and system design are matched.
- Storage requirements and auxiliary heat/energy sources are identified.
- Effective energy cost formula is applied and initial optimization is performed.
- Upgraded plant design aimed at optimal cost effectiveness is defined as baseline.

- Cost sensitivity analysis is performed to verify baseline performance. The parametric curves show optimum design is a function of economic tradeoffs; lifetime, money rates, tax rates, market projection, etc.
- Maintenance costs are included as a percent of capital investment.

Extension of life from 20 to 30 years can reduce equivalent energy costs by 15%, but the extension can be risky and in an unfavorable environment costs can increase 5% or more. Extended lifetimes may optimize earnings, but they do so at the risk of increased exposure to economic uncertainty. The 20-30 year framework appears, however, to be the most promising amortization range.

3.4.4 Mirror Field Sizing

The degree of solar dependence and optimum provision for fossil auxiliary heat is peculiar to each latitude, meterological region and insolation clime. Heliostat area (and, therefore, cost) is primarily influenced by three parameters:

- Average insolation level
- Solar energy required (absolute and percent of total)
- System efficiency.

Mathematically derived data, corrected for diffusion losses and cloudiness, agree well with recorded climatic data for Portsmouth, NH. Mathematically derived data assumed sun declination varying from 23.3^{o}S on 21 December to 23.3^{o}N on 21 June according to the expression:

$$d = 23.3 \cos \Theta \qquad (3\text{-}1)$$

The angle Θ is the season angle, or approximately the earth's position in its planetary orbit measured from 21 December. The expression was evaluated for 30° increments of Θ, and the sun's declination was taken as constant for the month. While the mathematic months are centered on the 21st day, the data (calendar) months are centered on the 15th day so a 6 degree phase angle is introduced with the real data year lagging by six days. The real data is plotted along with a cosine approximation (see Figure 3-14).

$$F_I = 5200 - 2400 \cos \Theta \ (kw/hr - m^2 da^1) \qquad (3\text{-}2)$$

The average real data lag is 12 days. Since six days lag is introduced by the different center points of the real and mathematic months described above, a real average lag of six days is attributable to climatic lag.

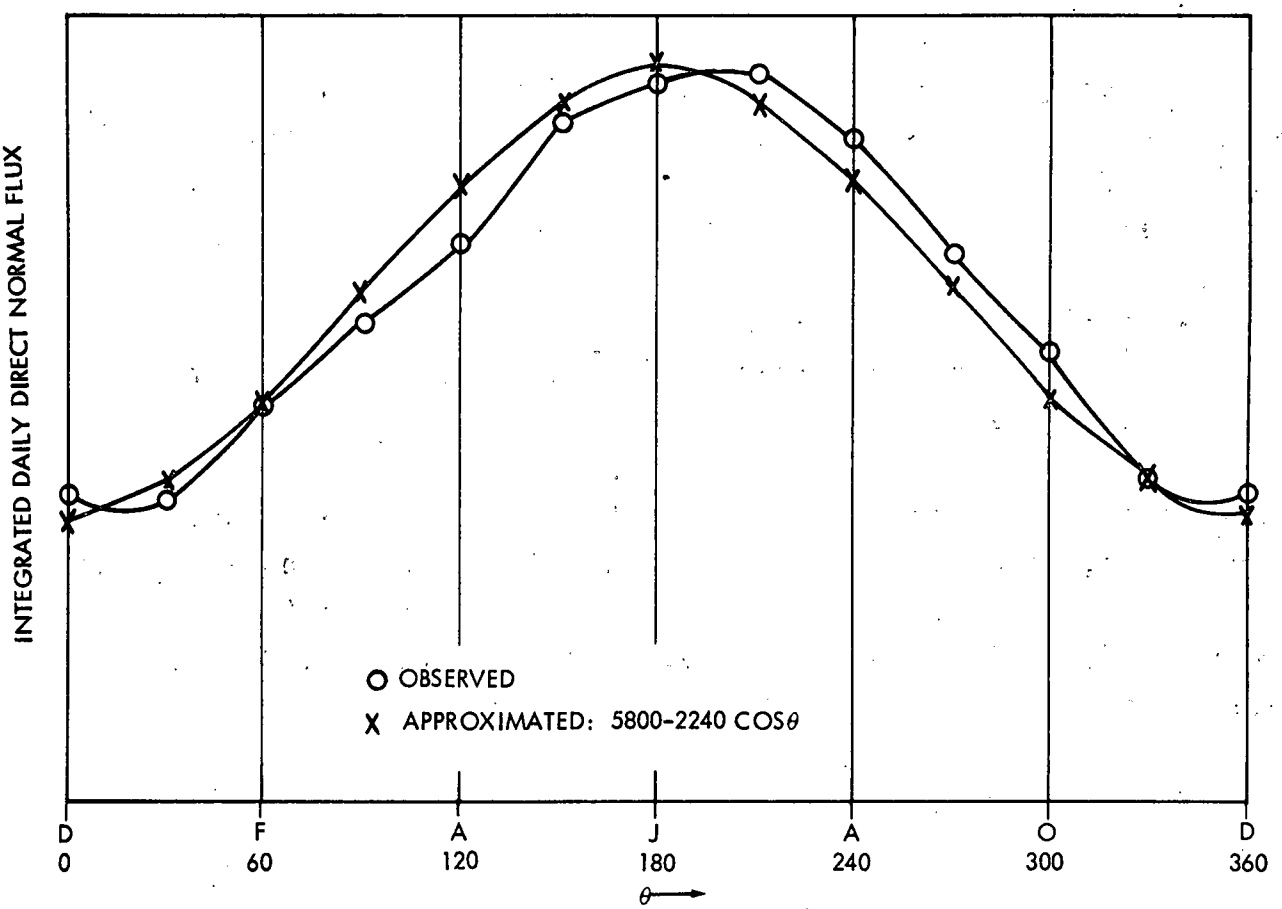


Figure 3-14. Daily Integrated Flux

The elevation of a celestial object is given by the expression

$$Hc = \sin^{-1} (\sin L \sin d + \cos L \cos D \cos LHA) \qquad (3\text{-}3)$$

where

Hc = elevation angle
L = observers latitude (N is +, S is -)
d = declination of object (N is +, S is -)
LHA = declination of object (N is +, S is -)

Neglecting atmospheric refraction, the LHA at sunrise or sunset (when Hc = 0.0) is given by

$$LHA_{RS} = \cos^{-1} (-\tan L \tan d) \qquad (3\text{-}4)$$

The average daytime sun elevation ($\overline{Hc}$) may be calculated by the expression

$$\overline{Hc} = \sin^{-1} \frac{LHA_{RS} \sin Hc \, dt}{2 \, LHA} \qquad (3\text{-}5)$$

The gain (csc $\overline{Hc}$) for a tracking normal surface compared to a horizontal surface may then be determined. Observed data for horizontal surfaces can then be converted to normal insolation values. The observed data includes weather-related losses from thin cirrus and thin obscuration to heavy overcast and precipitation. The integrated daily flux is based on a monthly average. Clear days would exceed the average and stormy days could well have no useable direct radiation.

The data of Moon of the Franklin Institute (1940) is plotted in Figure 3-15. It shows the normal direct flux on a clear day (East Cost of USA, 1940) as a function of elevation angle. The calculated direct insolation, at normal incidence, is determined by the expression

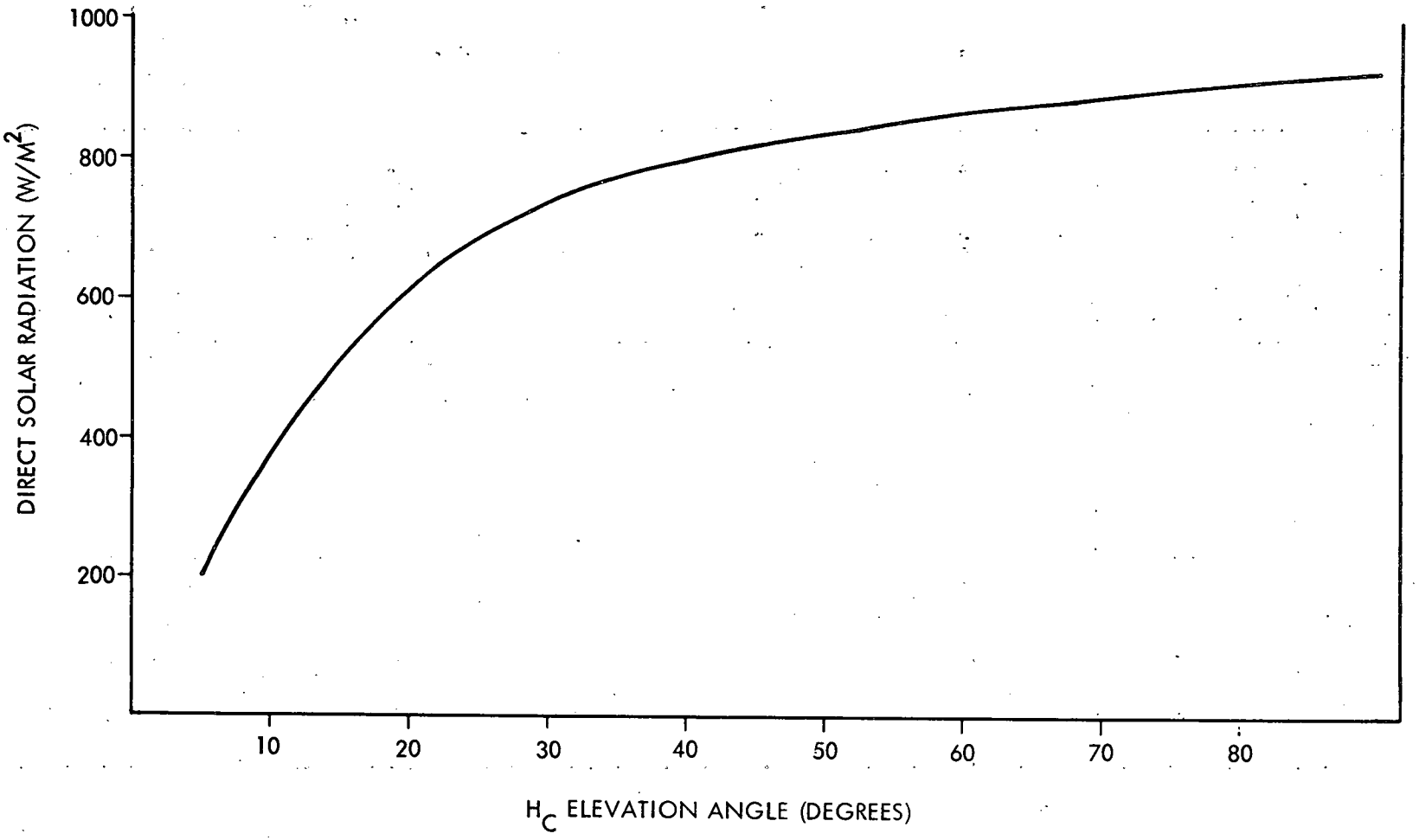


Figure 3-15. Normal Direct Flux on Clear Day.

$$I_D = \frac{2\ LHA_{RS}}{15} \times F_D, \overline{H}c \qquad (3\text{-}6)$$

where

I_D = direct insolation, at normal incidence, received during daylight hours.

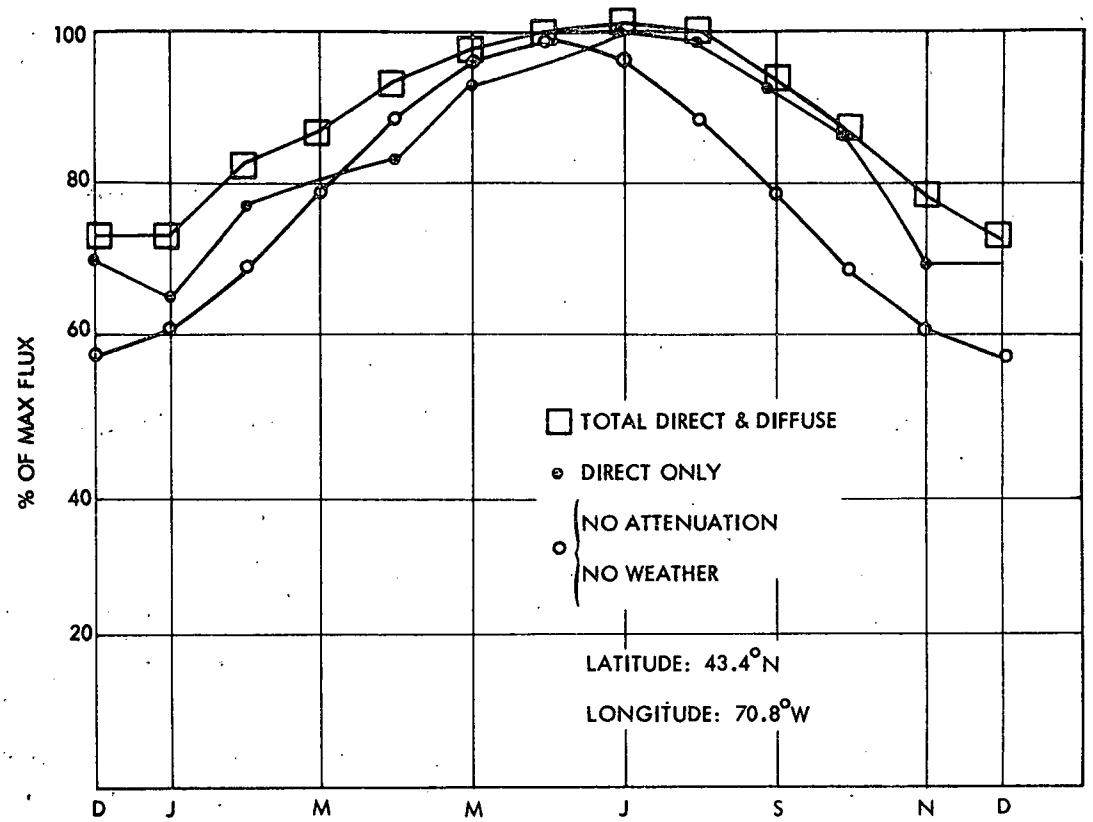
$\frac{2\ LHA_{RS}}{15}$ = duration of daylight in hours.

$F_D, \overline{H}c$ = direct flux, corresponding to average elevation angle, $\overline{H}c$.

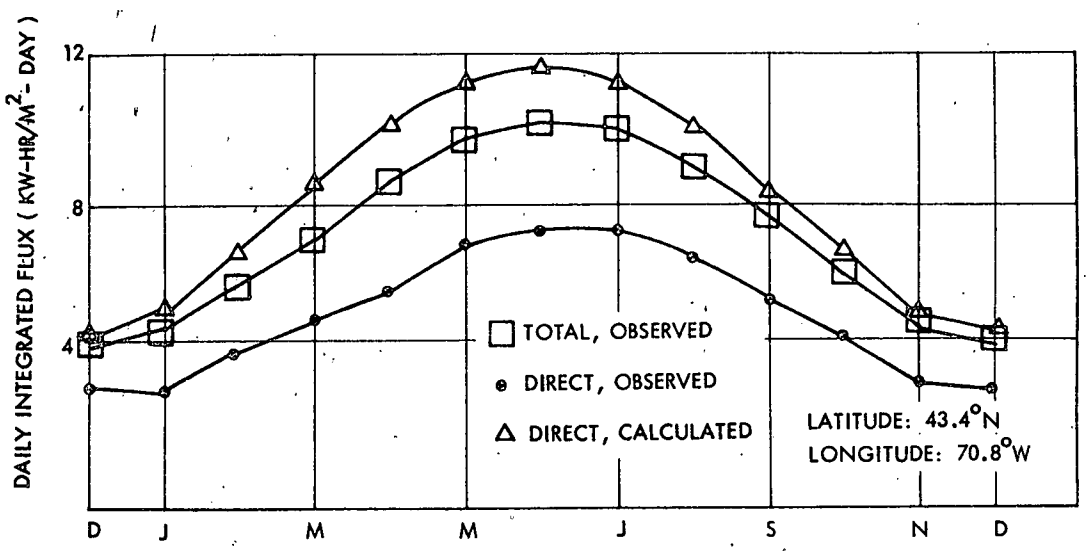
Figure 3-16 shows a plot of the daily integrated fluxes (calculated direct, observed total, and observed direct). The disparity between the calculated and observed direct radiation represents weather induced losses. Figure 3-16 also shows a time comparison of normalized fluxes (calculated total, observed total, and observed direct) as a percent of their respective maxima. Note these are not integrated fluxes (energy per unit area), but are rather fluxes (power per unit area).

Figure 3-17 shows the direct sunlight received as a fraction of direct sunlight possible and mean sky clearness. The apparent disparity between Air Force cloud cover information and weather bureau observations may be explained by the following:

(a) Micrometerological effects from the proximity of Great Bay to Peak AFB tend to increase observed sky cover during northwest circulations, which is generally the good weather direction, but the bay lies to the northwest of Pease AFB and clouds or fog may be swept into the base.



NORMAL SOLAR FLUX % OF MAXIMUM VS. SEASON



MONTH OF YEAR

DAILY NORMAL INTEGRATED FLUX - (W-H-M^{-2}) VS SEASON

Figure 3-16. Flux Values Versus Season.

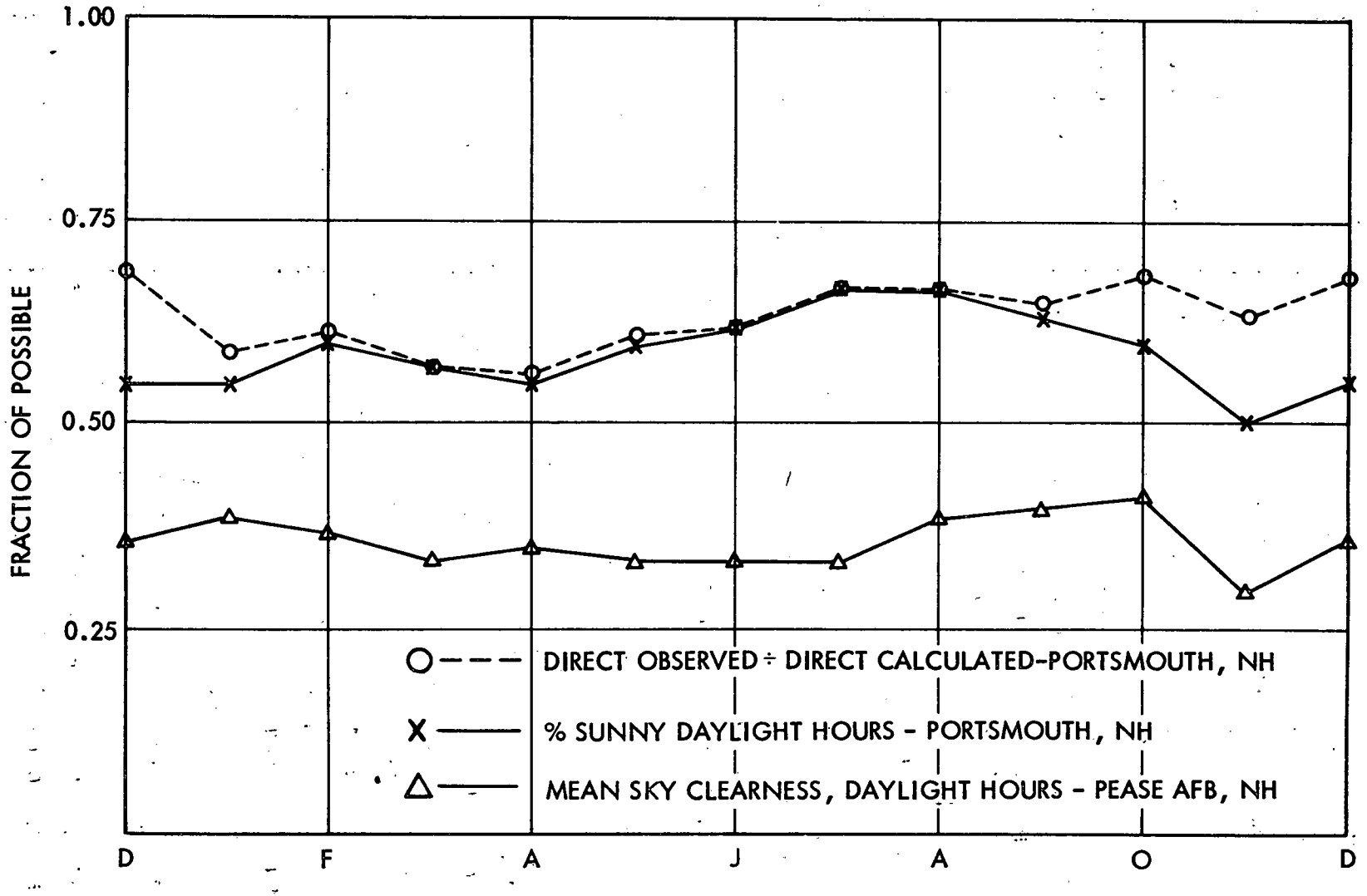


Figure 3-17. Direct Sunlight Received as a Fraction of Possible Sunlight and Mean Sky Clearness.

(b) Air Force cloud observers consider high cirrus a sky cover, but high thin cirrus not sky cover. The amount of sky cover is somewhat subjective upon the observer.

(c) Percent of possible sunshine is a statistic based on daily minutes of sunshine received as a fraction of the total possible (per month). The sunshine sensor is a photo cell device which can be activated by "bright" diffuse or direct sunshine. Clear bright sun does not reliably activate the sensor at low sun angles so here again a judgement factor is involved. In general, the percent sunny figures of recorded data are higher than the percent direct sunlight which actually falls.

(d) Low clouds or obscurations resulting from radiational cooling generally burn off in early to mid-morning hours. The sunny time lost during burn off is proportionally greater than the actual rate integrated insolation lost to the surface observer. During midday hours when the sun is at a higher elevation the direct radiation arrives at more than twice the rate it does near dawn and dusk.

The curves are believed to be generally consistent within their respective definitions. The received direct insolation data is used for plant sizing; this corresponds to the uppermost curve of Figure 3-17. In view of the definitions of the parameters plotted, the selection is considered safe, certainly not introducing an error of more than 10%. One must be careful, however, when siting the plant to assure micrometeorological phenomena will not invalidate the energy cost projections.

3.4.5 Power and Fuel Requirements

The economic design methodology led to the baseline system in which thermal-to-electric conversion is accomplished by a

modified Brayton cycle gas turbine generator. Modification of the cycle occurs in the heat addition phase. Heat is added to the working fluid (compressed) as it transits one of the storage tanks. Meanwhile the other tank is being heated by forced draft from either the solar receiver or the combustor. Hybridizing the plant saves hundreds of thousands of dollars in capital investment of heliostats and storage, reduces risk of adverse weather perturbations, and adds generally less than 1¢ per kw-hr produced. Hybridizing the system is best accomplished by providing alternate or complementary heat sources for the storage tanks. This technique permits the use of Number 6 fuel rather than forcing use of the more expensive light oils or jet turbine fuels. It also simplifies engine plumbing while only slightly impacting the storage system plumbing.

The plant, as sized and designed, will require 343 kw for pump and mechanical energy and 514 kw thermal energy. A total energy system can generate the needed electrical power and produce sufficient heat to satisfy the thermal requirements so long as thermal to electric conversion efficiency is equal to or less than 40%.

Using a cascade (overall) efficiency of 27.5% for a top cycle plant, heat is rejected at 204°C (400°F). Adding a bottoming cycle plant to boost overall efficiency to 31.6%, heat is rejected at 52°C (126°F). The rejected heat just matches the thermal load requirements. The mirror field can be sized to provide any selected fraction of plant requirements. The methodology used by Sanders was to size the collector field according to a percentage of the average annual requirements, and calculate the year's supplemental energy requirements. The most efficient (least costly) method of supplying supplemental energy was used to project the supplemental energy costs. If the supplemental energy costs are divided by the annual energy requirement, a distributed supplemental energy overhead rate can be computed.

Since the aquaculture plant requirement of 343 kw electrical is essentially constant and the solar insolation varies nearly as a cosine function (Equation 3-2), the power delivered is:

$$P = x - y \cos \Theta \tag{3-7}$$

and the energy deficiency at any time of year can be described mathematically as:

$$D = (x - y \cos \Theta) A - Z \tag{3-8}$$

where,

P = power delivered
D = power deficiency
x = mean flux
y = flux variation double amplitude
z = input radiation requirement (kw)

If during winter, the field supplies 60% of the requirements then:

$$x - y = 0.6 Z$$

An original rough cut of field size and associated tower geometry was sized and then analyzed using the Sanders mirror tracking computer program MTRACK. The effective mirror cosine factors are shown on the field mat of Figure 3-18. Cosine contours are plotted for the two extreme sun declinations. The grid coordinates are spaced at 31.15 meters (102 ft) intervals. Cosine values are accurate to three significant figures, integrated and averaged for the daylight hours. Contour lines are accurate to two significant figures, though in some cases a three figure nominal value is listed. Blocking effects were not calculated because the assumed packing density is only 39%.

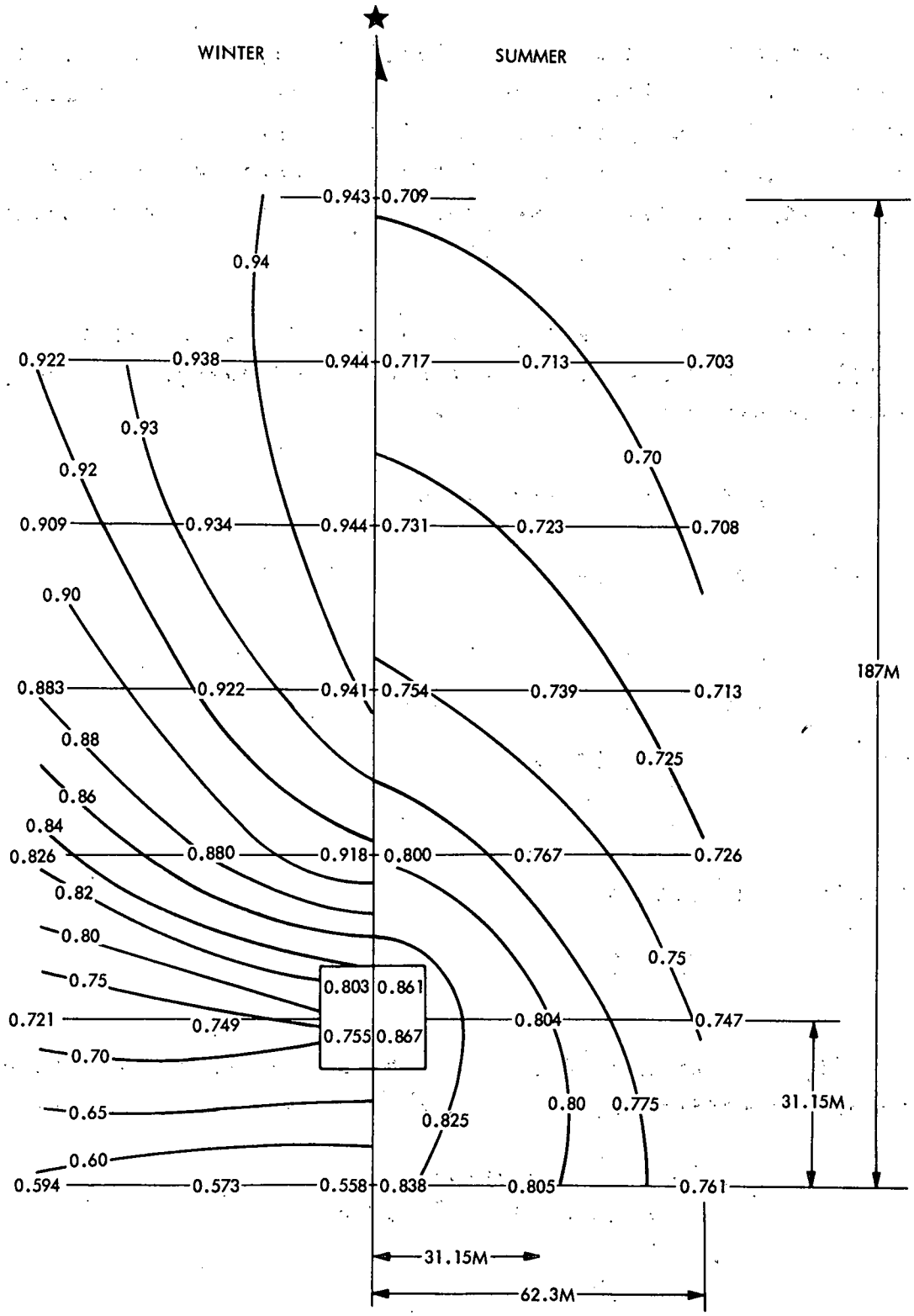


Figure 3-18. Aquaculture Mirror Field Daily Integrated Cosine Factors

Summer and winter redirected daily integrated fluxes and contours are similarly shown in Figure 3-19. The annotated field averages apply only to that portion of the field north of the tower. Examination of the winter map shows the north field reflecting to a south tower receiver optimizes performance at the latitude of interest (43.4^{o}N).

Referring to equations (3-7) and (3-8), the power delivered by a field which redirects half the required energy (per day) at the winter solstice is expressed by:

$$p = (x - y \cos \Theta) \text{ area} \tag{3-7}$$

Referring to Figure 3-19, it is obvious that

$$x - y = 2.76 \text{ kw-hr/da-m}^2$$

$$x + y = 5.45 \text{ kw-hr/da-m}^2$$

Thus, $x = 4.11 \text{ kw-hr/da-m}^2$

and $y = 1.35 \text{ kw-hr/da-m}^2$

For winter conditions,

$$A = \frac{P \cdot \text{time}}{2.76 \cdot \eta_\varepsilon}$$

If P is taken as 206 kw at the winter solstice, and overall conversion efficiency $\eta_\varepsilon = 0.316$, then the required field area is

$$A = \frac{206 \text{ kw} \times 25 \text{ hr/da}^{-1}}{2.76 \text{ kw}\cdot\text{hr}\cdot\text{da}^{-1}\text{m}^{-2} \; .316}$$

$$= 5670 \text{ m}^2$$

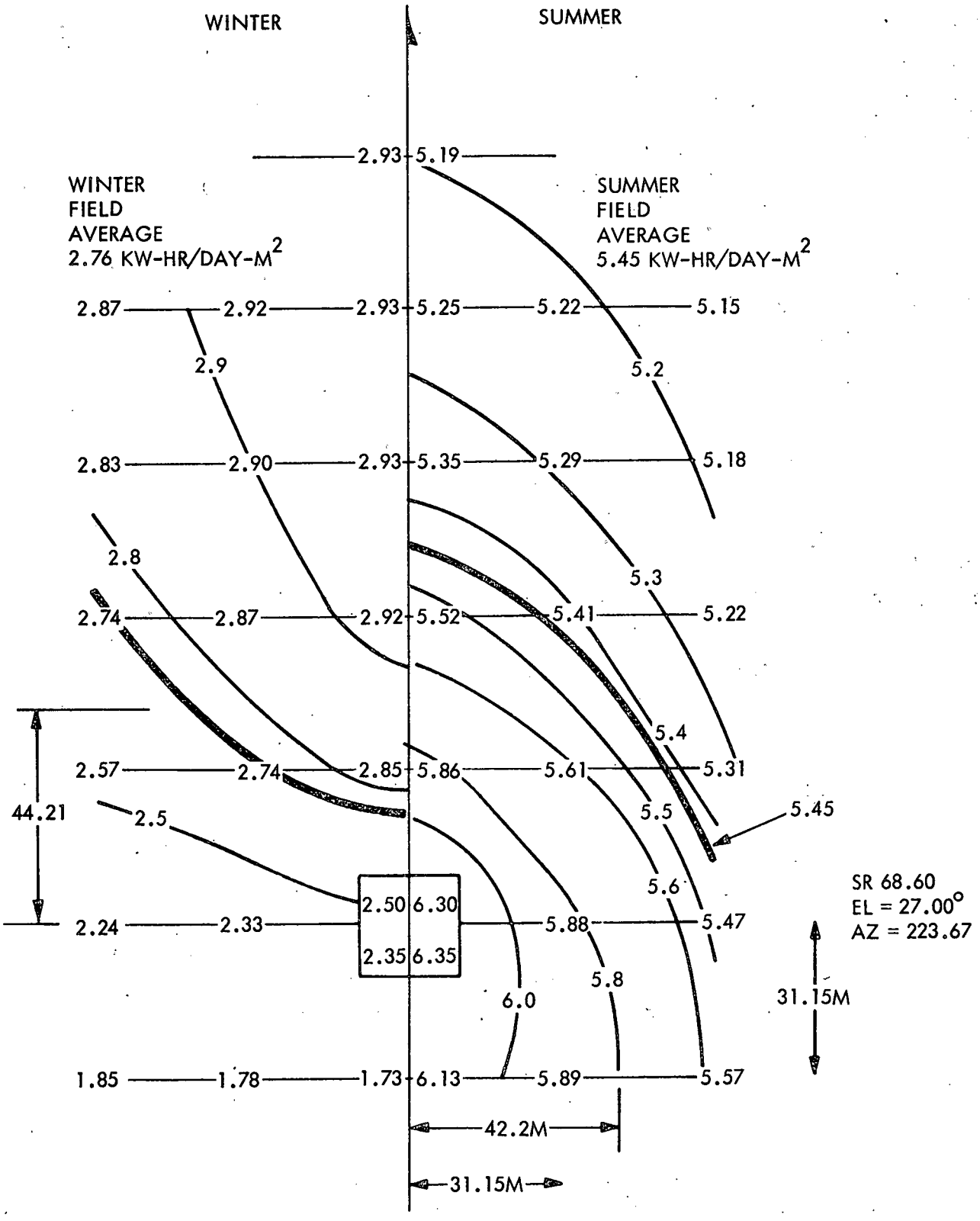


Figure 3-19. Aquaculture Mirror Field Redirected Daily Integrated Flux -(100% Reflectivity)

A solar energy deficiency exists for some period on both sides of the solstice. When solar insolation just matches requirements, the deficiency is zero and equation (3-8) may be rewritten to solve for Θ:

$$\Theta = \cos^{-1}\left(\frac{Az - z}{Ay}\right)$$

In the example above where A is 1790 m^2 and z is 343 kw:

$$\Theta = \cos^{-1}\left(\frac{5670 \cdot 4.11 \cdot .316 - 3.43 \cdot 24}{5670 \cdot 1.35 \cdot .316}\right)$$

$$= \pm 1.94 \text{ radians}$$

The plant, which will provide 60% of the requirements from solar at the solstice, will require supplemental fuel for 226 days per year. The total supplemental energy required is equal to the solar deficiency (3-8) integrated over the shortfall period:

$$E = \int D d\Theta$$

$$= \int (xA - Z) dt - \int \frac{yA}{w} \cos \Theta d\Theta$$

$$= \int_{-113}^{113} \left(4.11 \cdot 5670 - \frac{343 \cdot 24}{.316}\right) dt - \int_{-1.94}^{1.94} \frac{1.35 \cdot 5670}{5.476E-3} \cos \Theta d\Theta$$

$$= -2747 \int_{-113}^{113} dt - 1.398E6 \int_{-1.94}^{1.94} \cos \Theta d\Theta$$

$$= 6.208 \times 10^5 - 2.608 \times 10^6$$

$$= 3.228 \times 10^6 \text{ kw-hr}$$

This energy may be purchase either in a high grade form (all electric), in a mix matched to the load, or in fossil form. The results of these options are shown in Table 3-8.

TABLE 3-8

SUPPLEMENTAL ENERGY OPTIONS

Option	All Electric	Electric Power Fossil Fuel Heat	Total Energy System
High Grade/Electric	$3.22X10^6$ kwh	$1.020X10^6$ kwh	—
	$96,840	$30,600	
Low Grade/Thermal	—	$2.208X10^6$ kwh	$3.22X20^6$ kwh
			$28,410
Yearly Cost	$96,840	$50,030	$28,410
Cost/kw-hr(elect)	$.0300	$.0155	$.0088
Cost/yr / Total Energy $/kwh	$.01018	$.005258	$.002986

Use of the total energy system approach to provide supplemental energy of the quality required saves over 70% of the supplemental energy cost. This approach also fully utilizes the turbine generator sets to reduce the amortized capital cost per kilowatt-hour.

Fuel price inflation will increase the real cost of supplemental energy only insofar as that inflation exceeds the inflation rate of the general economy. In other words, with supplemental energy set at $0.088 per kilowatt hour the 1980 cost deinflated to 1977 dollars would be

$$\$/kwh = .0088 \left[\frac{1 + \frac{i_{fuel}}{100n}}{1 + \frac{i_{economic}}{100n}} \right]^{nt}$$

where, i_{fuel} and $i_{economic}$ are averaged inflation percent rates over the interval, n is number of periods per year and t is number of years.

Maintenance expenditures are of three types: Preventive, scheduled, and unscheduled. For planning purposes the maintenance budget is set as a fixed percentage of original plant capital expenditures. By providing a fixed percentage in excess of the preventive and scheduled budget, a reserve can be provided for unscheduled maintenance. The maintenance budgets are reasonably linked to the inflation rate so the cost in first year dollars remains constant over the plant lifetime.

3.4.6 Capital Investment

The size of the mirror field in effect determines the capital investment. Mirror field size is related directly to the preparation of the total energy budget which is supplied by solar through its weather data presented in Appendix B for Portsmouth, NH, and the efficiencies presented in Table 3-7 and Figure 3-20. Also, presented is the proportion of energy supplied in December when insolation and sunny days are at a minimum. For example, a mirror area of 55,000/ft^2 will theoretically supply 100% of the energy needs for the year, but only 60% of the December demand. Obviously, fossil fuel must be used to makeup the deficit. The tower height is presented in the upper graph and reaches 85 ft for the example above.

The storage requirement, Figure 3-21, varies in this study according to the proportion of energy collected from solar. Again for the example of 100% of the yearly demand coming from solar, the storage volume will provide 27 hours of operation and will require 580 tons of firebrick. Smaller mirror fields will have shorter periods of operation from storage.

Capital investment is the sum of the cost of power plant, storage stoves, receiver mirror field and control mechanims, and tower. All these cost elements except the power plant will increase with an increase in the percent of solar energy collected as

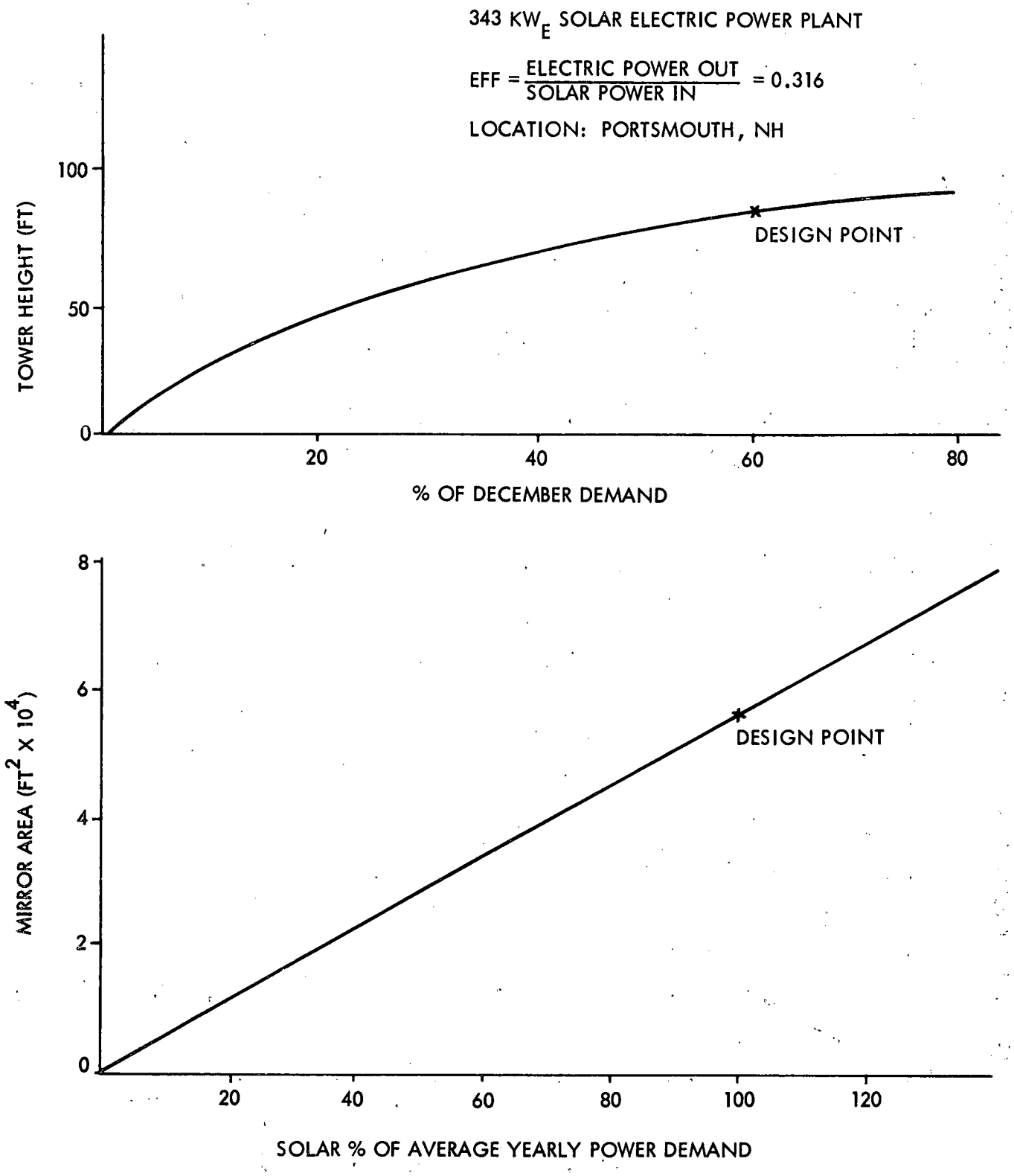


Figure 3-20. Power System Geometry versus Power Demand.

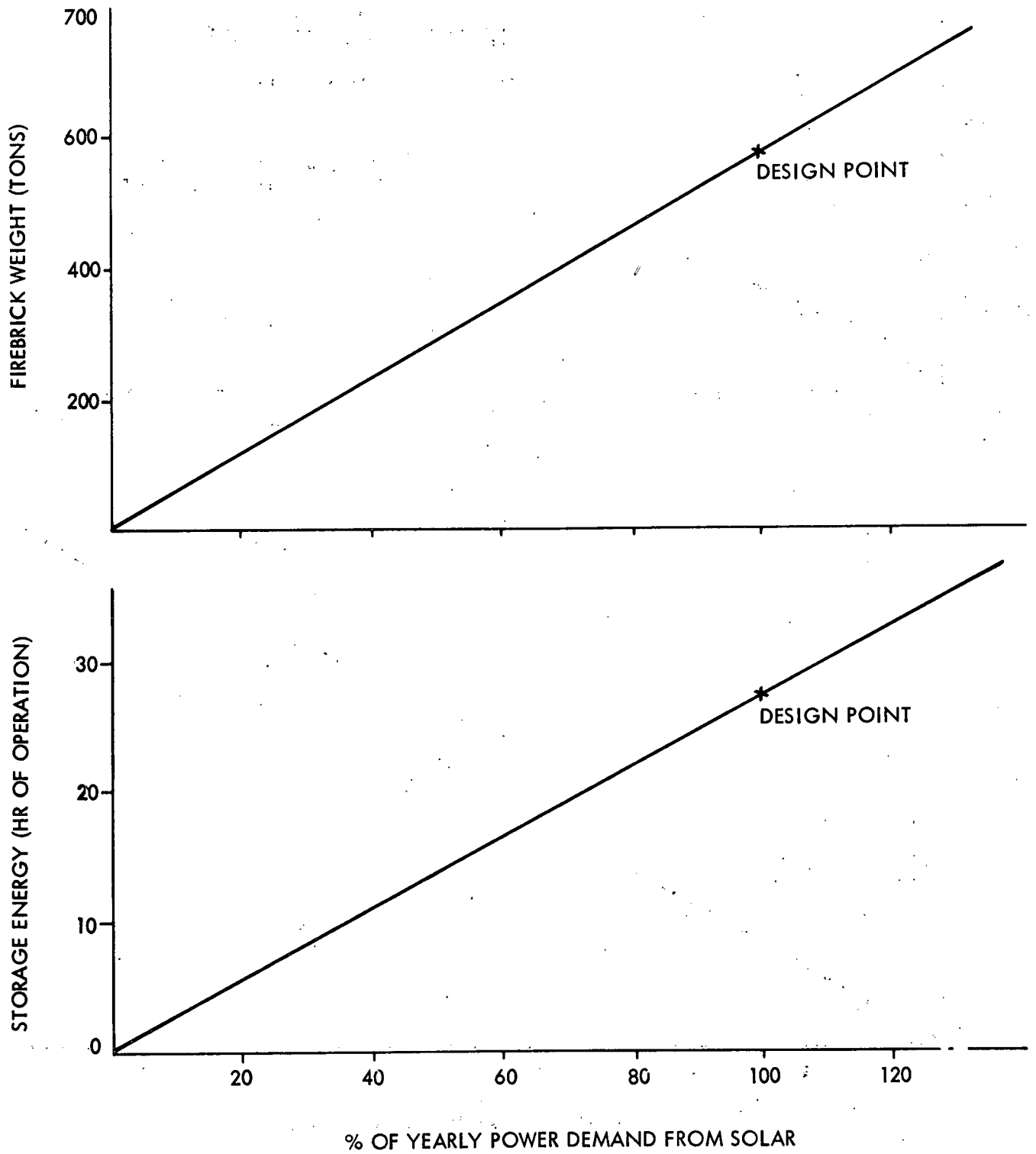


Figure 3-21. Storage Requirements versus Power Demand.

shown in Figure 3-22. For a plant collecting 70% of the yearly demand from the solar system, the capital investment is approximately 1 million dollars in todays dollars.

3.4.7 Equivalent Cost

The objective of the equivalent cost analysis is to first determine the point where solar energy investment will save money over the present cost of electric power and oil heat, and secondly, determine the size plant which is most cost effective. The analyses will include depreciation periods up to 30 years.

Power generated and fuel cost used in its analysis are shown as a function of years in Figure 3-22. Note that the fuel costs start at $.008/kwh and rise to $.014/kwh after 30 years just due to the 10% per year fuel inflation rate.

As a reference point in the analysis, two equivalent cost curves were developed covering the depreciation period to 30 years, Figure 3-24. The curve shows the equivalent fuel cost for a plant using all electric power. Equivalent cost of electric heat starts at $0.030/kwh and rises because of inflation to $0.034/kwh in 20 years.

From this analysis, the optimum design for a 20 year depreciation period is one designed to meet 100% of the average energy demand per year (60% of December demand) at a cost of 1.3 million dollars. For shorter depreciation periods, a smaller investment is justified. The optimum design for any given depreciation period will be dependent primarily on the difference in fuel inflation rate with respect to interest rate and economic inflation rate. Even the total energy plant without solar energy is cost effective compared with electric power and oil heat over a 10 year depreciation period.

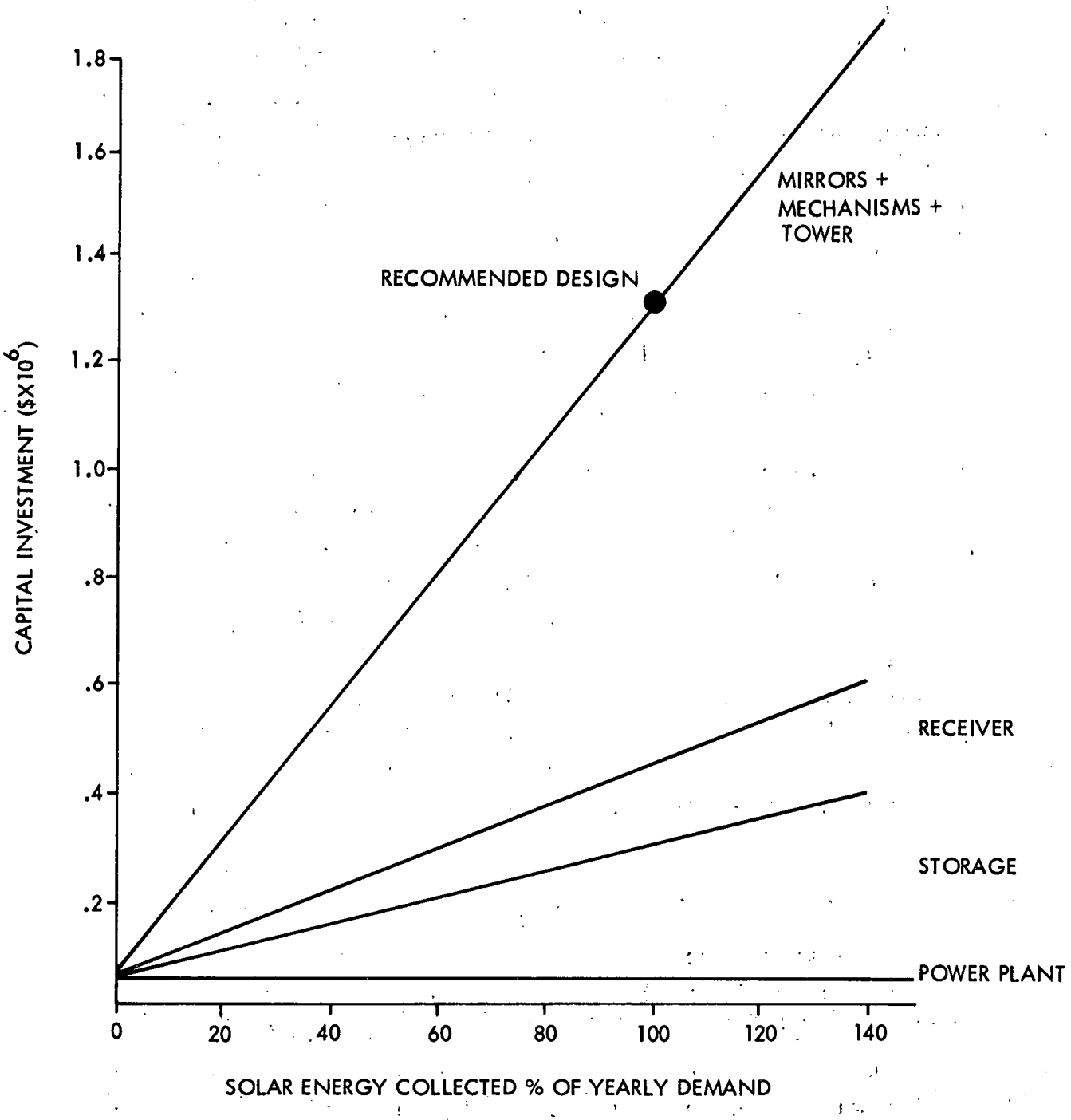


Figure 3-22. Capital Investment versus Power Demand.

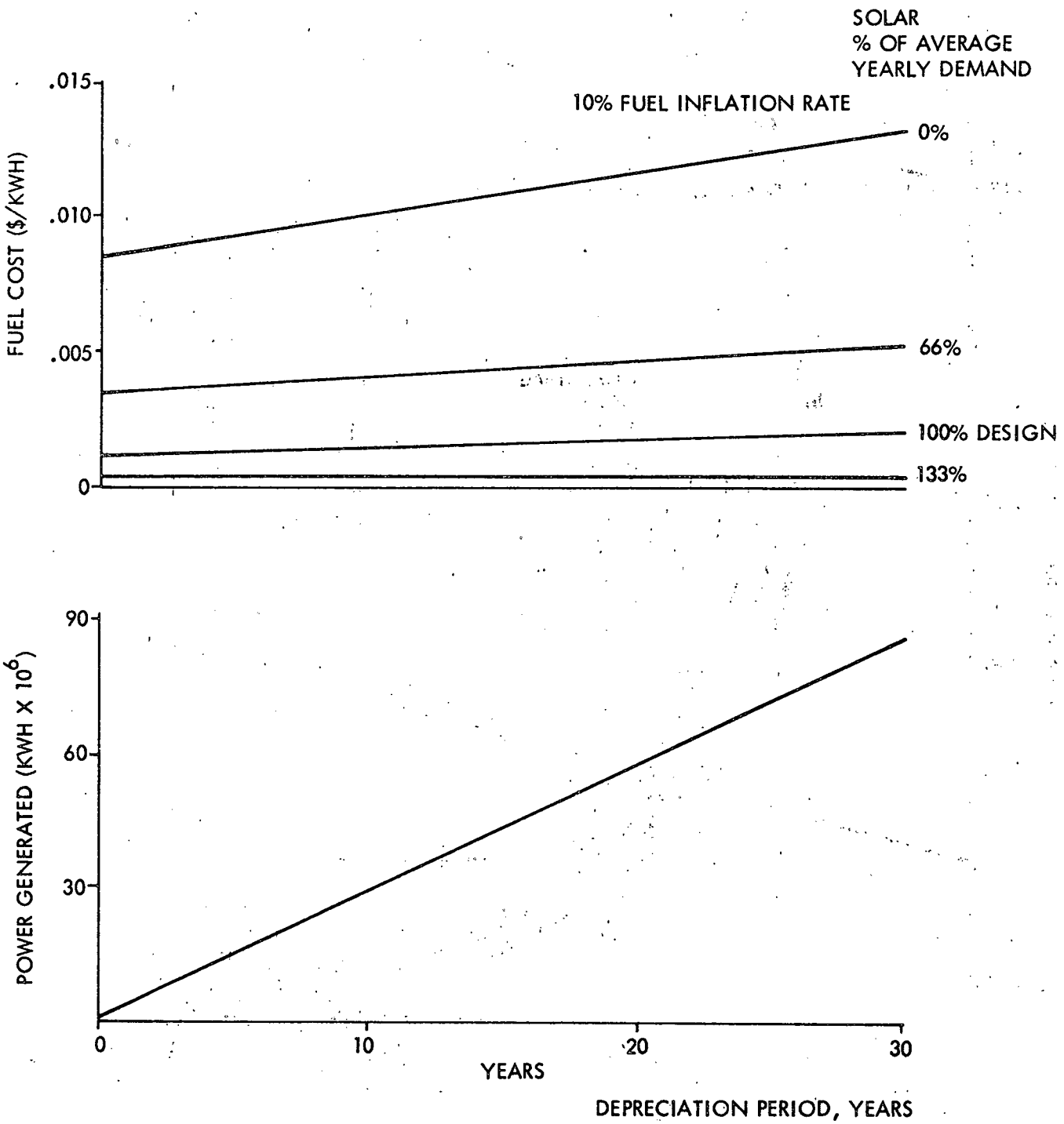


Figure 3-23. Power and Fuel Requirements.

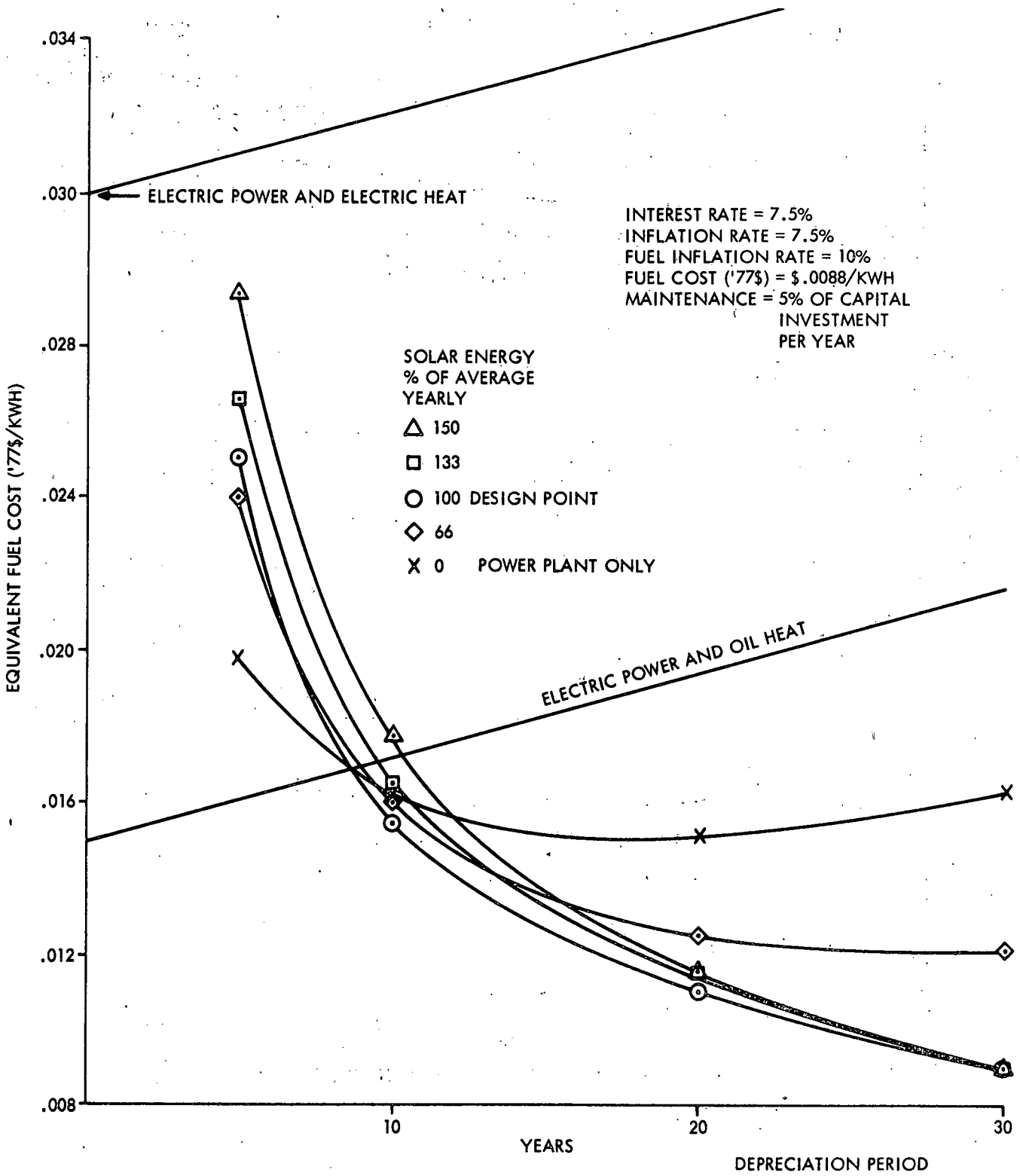


Figure 3-24. Aquaculture Solar Power Cost Analysis.

3.5 AQUACULTURE TOTAL ENERGY PLANT

3.5.1 Production Plant Layout

The production plant capable of marketing 800,000 lobsters per year consists of a main production plant containing 130,000 square feet. A solar energy power plant adjacent to the building Figure 3-25, occupies 3.5 acres of cleared land. On the south side of the mirror field will be an 85 ft tower containing:

- Receiver assembly
- Three storage stoves (27 hours of operation)
- Two gas turbine-generators
- Organic Rankine engine generator
- Control system
- Transformer, fans, valves and instrumentation

Within the mirror field are 405 individually directed mirror mounts containing both azimuth and elevation controls for the 4.2 m Dia mirror. The system block diagram (Figure 3-26) shows all the power and heat coming from the total integrated power plant. The storage stoves contain the heat required to drive the alternators. This heat can be supplied by either a fossil fuel combustor or solar energy.

Three storage stoves are built into the power tower and can function in one of three modes:

(1) Receive and store solar heat from mirror field - ambient pressure.

(2) Receive and store fossil fuel heat.

(3) Discharge heat to gas turbine generator at three atmospheres.

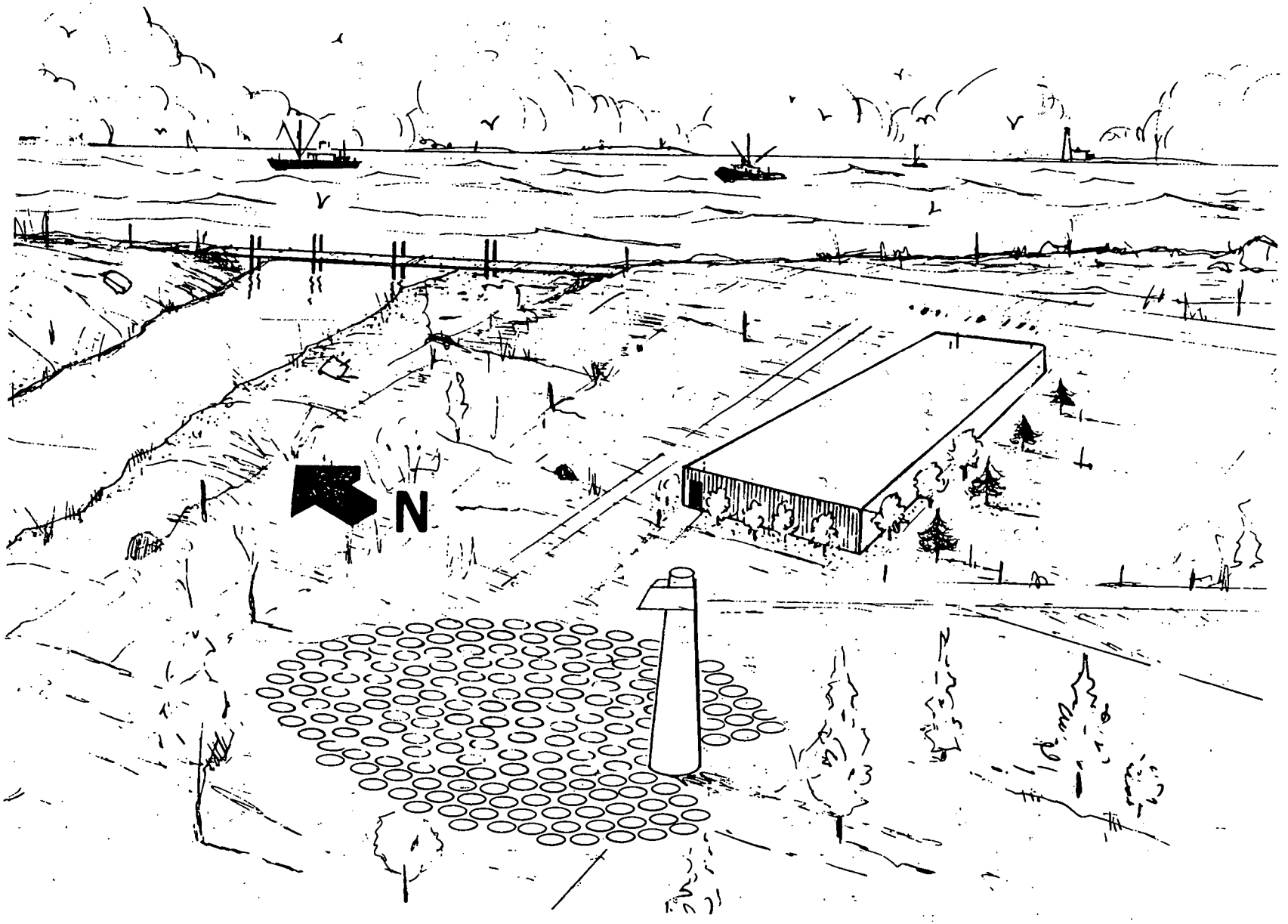


Figure 3-25. Aquaculture Total Energy Plant with Solar Power.

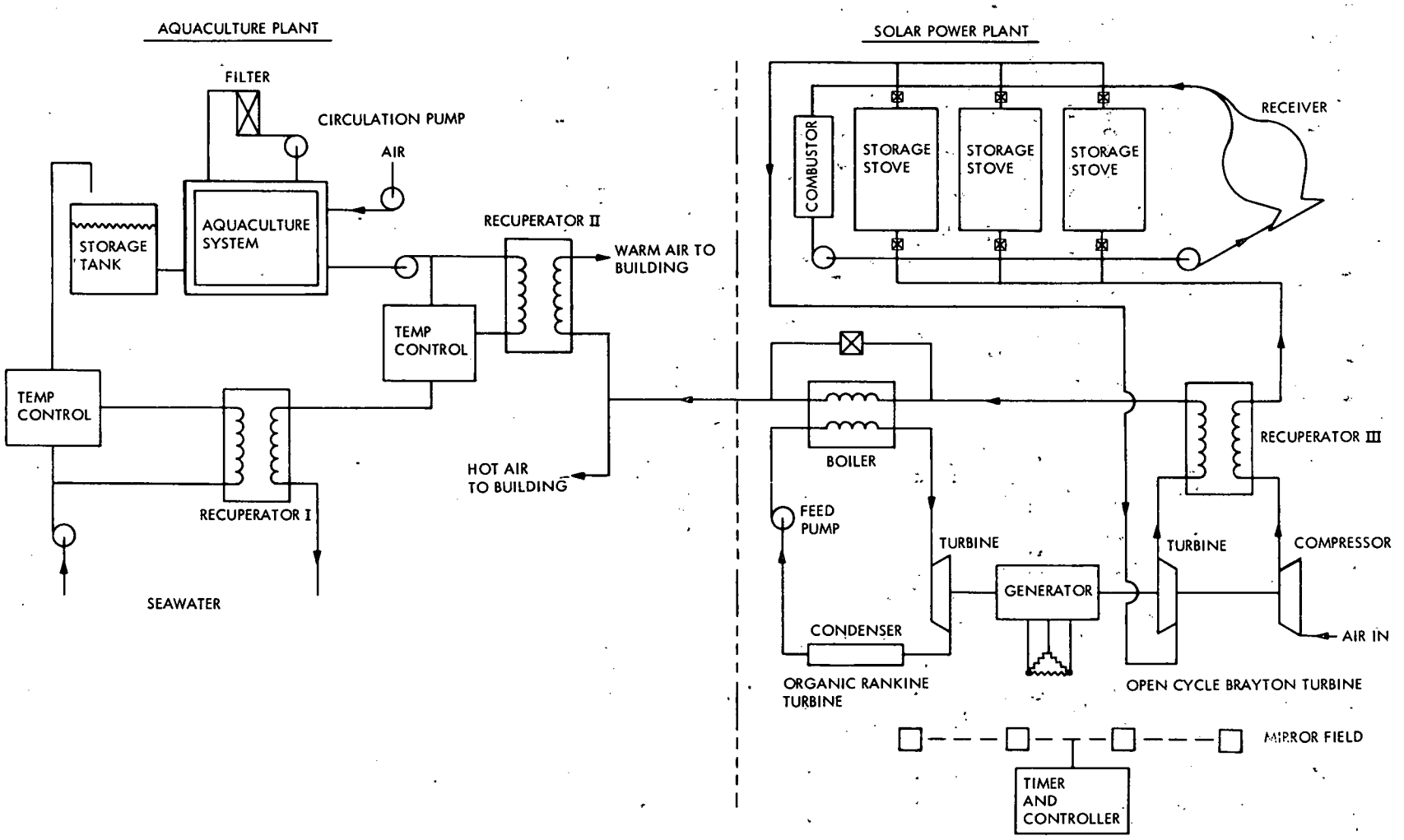


Figure 3-26. Block Diagram - Solar Total Energy Aquaculture Plant

The capital investment, exclusive of land and utilities, is estimated at $\$9.3 \times 10^6$ for the production plant and $\$1.3 \times 10^6$ for the total energy power plant.

3.5.2 Instrumentation and Control

Instrumentation and controls are located in the aquaculture production plant for those functions concerning the biological growing process another station in the base of the power tower contains instruments and controls for those functions which are needed in generating electric power. From Figure 3-26, aquaculture production controls are needed to supply nutrient-rich seawater at $68^{o}F$ and the same value for the plant air temperature. These functions are controlled by regulating the air flow into the building and through recuperator II and the water flow through recuperator I.

Automatic operation of the solar power plant is provided by a mini-computer which matches power output to demand, and provides the necessary switching between storage stoves, compressors, turbines, boilers and recuperators. A separate control system generates the signals which independently calibrate and control the mirror tracking of each assembly.

SECTION 4
CONCLUSIONS

Under a joint program between ERDA and Sanders, solar energy was examined, in concept and cost, to assess its effectiveness in reducing the operating cost of a lobster aquaculture plant located in Kittery, Maine. Using the Sanders solar energy pilot plant as a model, thermal loads and auxiliary power requirements were estimated for a full-scale commercial plant capable of producing 800,000 lobsters annually.

The results of this analysis indicated that plant operation using the heat from flat plate collectors would not be cost effective. Organic Rankine engines were also considered and were rejected because of low efficiency. A total energy power plant having a concentration ratio of 1500 to 2000 and delivering both heat and electric power would be cost effective compared with conventional energy costs when the capital investment is amortized over ten years or more.

A total energy system design concept was developed with specifications and costs for the recommended plant situated either on the Piscataqua River or on an island off the Maine coast. The general arrangement of components and the necessary instrumentation was presented for the recommemded full-scale commercial plant.

The recommended power plant needed to operate a 130,000 square foot lobster production facility was designed to provide 343 kw of electric power on a continuous basis from solar energy. This is 100% of the average yearly power demand. The plant would also provide all the required building and water heating. Because of the lack of sunshine during the winter months, a fossil fuel combustor would supplement the solar energy to the extent of approximately 40% of the December demand. This plant would consists of 60,000 square feet of mirror surface, spaced to minimize mutual inteference, over a 3.5 acre elliptical area. Each mirror would have individual tracking

such that 1500 to 2000 solar images would be concentrated on the solar receiver located on an 85 foot tower in the southern section of the mirror field. Three sensible heat storage stoves would be contained in the tower and would be capable of operating a Brayton gas turbine generator for 28 hours. All the power generating equipment and controls would also be contained in the tower.

Power plant costs were estimated at $1.3 million dollars. A 30 year amortization period was assumed with 3×10^6 kw-hr of electricity produced annually. A cost analysis, which assumed an inflation rate of 7.5%, an interest rate of 7.5%, a maintenance cost of 5% of capital investments per year, and a fixed inflation rate of 10% per year, resulted in an equivalent fuel cost of $0.0088/kw-hr. This is compared with the present commercial rate of $0.015/kw-hr for electric power and oil heat.

APPENDIX A
THE LOBSTER AQUACULTURE PRODUCTION PLANT

A.1 INTRODUCTION AND BACKGROUND

Aquaculture, the culture or husbandry of aquatic animals or plants, has been known in one form or another for over 2,000 years. Until recently natural stocks of marine resources were thought to be unlimited, but are now considered to be capable of sustaining a maximum harvest of 100 - 150 million metric tons (220 - 330 billion pounds) per year. On a worldwide basis, a shortage of fish products is emminent within 10 years due to a combination of population increase, depletion of natural fish stocks, and the sky-rocketing costs of energy and its impact on agricultural processes.

A.1.1 Importance of Aquaculture

In the United States, most of the traditional fish resources are harvested at or near maximum sustainable yield levels. Imports have increased, but world demand has also expanded. This situation will limit the amount of seafood available to the United States from outside sources, causing this commodity to be excessively expensive. Thus, the United States' demand for traditional seafoods will become critical within the next decade, resulting in supply shortages and increased prices of many products.

Worldwide output from aquaculture has approximately doubled during the last five years and now totals nearly 6 million metric tons (13.2 billion pounds), roughly 10% of world fish production. Some countries already rely on aquaculture for over 40% of their total fish supply and expect this to increase.

In the United States aquaculture has been, until recently, lagging in activity compared to many other countries. More than one quarter of United States' salmon (27,000 metric tons or 60 million pounds) originates in hatcheries. Private aquaculture produces 40% of the oysters, half of the catfish and crayfish, nearly all of the trout, and small quantities of several other species for a total of 65,000 metric tons annually (143 million pounds). However, this weight accounts for only 3% of U.S. catches and 2% of U.S. total consumption of fishery products.

To provide new inertia to the lagging exploitation of aquaculture in the United States, the National Marine Fisheries Service of the National Oceanic and Atmospheric Administration (Department of Commerce) developed a National Aquaculture Plan. In this plan, species have been prioritized based on the health value of the natural fishery and the economic importance of the product. The species identified as being of highest national priority for cultivation based on these considerations are salmon, marine shrimp, fresh water prawn, American lobster, oysters and marine plants. Of these, Sanders chose the American lobster as the primary focal point for aquaculture development because of its high value as a product, its unique challenge from a system technology standpoint, and the applicability of resulting culture technology to other species of interest.

A.1.2 Background of Growth Cycle of the American Lobster in the Natural Environment

Lobsters are true products of their environment. Their growth, frequency of molt, and each phase of their life cycle depends on many factors that vary with location and season. Two lobsters may weigh the same, but one taken from the rocky coast of New Brunswick and the other from a wreck off the New Jersey shore are unlikely to be the same age. The life history of a lobster living in shallow water (15 ft) in Long Island Sound will unlikely be the same as that of one living in 1000 ft of water in the submarine canyons on the edge of the continental shelf.

Water temperature and food are the two most important factors affecting a lobster's growth and weight gain. But also important are: water salinity, availability of shelter, frequency of regeneration (replacement of lost appendages) and ocean floor condition. Lobsters grow much faster in warm water than in cold, and thrive when all the necessary food is found in a small search area.

In the summer months (between June and September) as water temperature reaches a maximum, the female lobster extrudes her eggs and cements them under her tail. Within her body, she carries the sperm of a 9 - 12 month's earlier mating which now fertilizes 3,000 - 100,000 eggs, depending on her age. This "berried" female as she is now called, will carry these eggs externally for an additional 9 - 12 months until they are mature and ready to hatch, a total gestation period of 18 - 24 months.

The following spring, when the eggs are matured, and largely dependent on the ambient water temperature reaching the appropriate levels, the female lobster will face into the current, raise her tail high and beat her swimmerets strongly; releasing clouds of new born and hatching lobsters over a period of several nights until all have dispersed. Initially the animals rise to the surface and feed on plankton brought there by upwelling and currents. In two to three weeks, however, after undergoing four or five molts, the young lobsters who have eluded natural predators and survived (0.2%) will sink to the bottom to feed on benthic marine organisms for the remainder of their lives.

A lobster normally takes 7 - 9 years after hatching to reach a "market" size of 0.9 - 1.0 lb, and is caught and released up to 10 times before it has reached legal size and can be marketed. Many are injured and many die during this rigorous existence, further complicating efforts to regulate and protect what is fast becoming an endangered species.

A.1.3 Advantages of Growth at a Uniform Temperature

A culture environment offers the American lobster many advantages. In order of importance these advantages are:

- Protection from stress and predation
- A supply of ample and nutritious food
- Control of natural disease

To economically accomplish this, the growth time to market size must be reduced drastically. Since the American lobster is a cold-blooded animal and derives its body temperature from its environment, the rate at which it can metabolize food and build body tissue is directly related to the temperature of the environment. Indeed, in some geographic areas where water temperatures remain low year round, natural lobsters rarely grow to over 3/4 lb.

A temperature range of 20° - $22^{o}C$ has been determined by researchers as the ideal biological temperature for optimum growth of the American lobster. At temperatures above $22^{o}C$, the animals often exhibit metabolic disturbances (coupled with reduced resistance to disease and stress). At temperatures below $20^{o}C$, the growth rate decreases. However, Sanders has determined from its experimental program that $18^{o}C$ is a more practical aquaculture system operating temperature due to the substantial increase of bacterial growth and deterioration of process water quality in the 20° - $22^{o}C$ range. Excursions to $14^{o}C$ are normal and expected in a system of this type. Significant gains in growth time can be realized, with 2 1/2 years already achieved compared to the 7-9 years in nature to produce a legal size lobster (approximately 0.9 lb). Other species experience

similar gains. The relative importance of elevated temperatures in the culture of the species primitized in the National Aquaculture Plan is depicted in Table A-1.

A.1.4 Contributions Which Can Be Made by Solar

Since the American lobster (Homaris Americanis) is native only to northern waters (New England, the Maritime Provinces and the North Sea), it can be assumed that these areas provide nutrient-bearing waters unique to the needs of Homaris. For this reason, it is appropriate that culture facilities be located on waterfront properties for the foreseeable future.

Large quantities of natural seawater are required for the initial filling of the lobster aquaculture facility and the required make-up water (17,000 gal/hr) to insure proper water quality. Hence, in order to raise the natural seawater temperature to a comfortable temperature which will promote more rapid animal growth, large amounts of thermal energy are needed. In addition, a continuous supply of high grade energy is necessary for pumping, aeration, filtration, and other materials handling and process needs.

A.2 THE LOBSTER AQUACULTURE PLANT

A.2.1 Introduction

Sanders chose to develop the industrial technology for the breeding and rearing of the American lobster after reviewing the entire aquaculture field. The American or northern lobster is traditionally a highly prized seafood. The U.S. market is seriously limited by supply and some authorities consider Homaris an endangered species.

TABLE A-1

IMPORTANCE OF THERMAL INPUT TO CULTIVATION OF SPECIES IDENTIFIED IN NOAA AQUACULTURE PLAN

	Critical To	Benefits By	Not Important
High-priority species			
Salmon		X	
Marine shrimp	X		
Freshwater prawn	X		
American lobster	X		
Oysters		X	
Marine plants		X	
Medium-priority species			
Low-cost fish		X	
Catfish		X	
Yellow perch	X		
Clams	X		
Abalone			X
Bay scallop		X	
Spot prawn		X	
Tropical shellfish	X		
Low-priority species			
Marine fishes			
Pompano			X
Sablefish or black cod			X
Striped bass			X
Tropical aquarium fish		X	
Tropical food fish			X
Tuna Baitfish			X
Trout		X	
Crawfish		X	
Mussel		X	
Crab	X		
Marine baitworms			X

U.S. landings of 28 million pounds (12,700 metric tons) were supplemented by imports of 18 million pounds (8,200 metric tons) in 1974. The National Plan for Marine Fisheries of 1976 projects a shortfall of supply versus demand of 40 million pounds (18,000 metric tons) in the United States by 1985; prospects are poor for increasing imports to provide this quantity. Offshore stocks have declined from their virgin condition and are considered irreplaceable by many authorities. The development of a technically sound and economically viable aquaculture system will help to alleviate the projected shortage, both by providing product to the marketplace and post-larval animals for stocking programs.

A.2.2 Lobster Aquaculture Background

Since 1950, the Massachusetts State Lobster Hatchery has reared larvae lobsters for release each year in various areas of the Commonwealth. Annual production has averaged 250,000 fourth-stage lobsters. Unfortunately, it has not been possible to release large quantities of these lobsters in any one particular area, and evidence is lacking that the release of larvae in the natural environment significantly improves local lobster fisheries.

The greatest technical contribution of the Massachusetts hatchery was the early evaluation of potential culture techniques which provide a base for aquaculture development. The American lobster has been cultured through all stages of its life cycle at this station, and it was found that larval development and growth rate of juveniles could be expedited by increased water temperature. Mr. John Hughes, manager of the Massachusetts State Hatchery, is an ongoing consultant to Sanders aquaculture program.

An $8^{o}C$ elevation in temperature can reduce the larval period by a factor of nearly four. Growth to U.S. market size, which requires 7-9 years in nature, can be reduced to two years if water temperatures are increased to about $21^{o}C$.

The potential for private aquaculture of lobsters has led to expansion of lobster research in other areas as part of a national program. The NOAA (Department of Commerce) Sea Grant program has included projects to develop lobster feeds (Louisiana State University and University of California); to determine nutritional, physiological, and environmental requirments for mass culture (University of California-Davis); to determine the potential of culture in thermal effluent from power plants (San Diego State University); to determine the efficiency of alternate methods for mass hatching and culture of larvae (University of Rhode Island and State University of New York); and to identify and isolate natural chemical substances which determine lobster behavior (Woods Hole Oceanographic Institute).

A.2.3 Sanders Associates Industrial Development Efforts

The major problem areas in the development of a commercially-viable lobster aquaculture system (identified by Sanders for proprietary in-house effort) include:

- Feeds
- Culture System
- Economics
- Waste Disposal
- Genetic Improvement
- Water Quality Management

The problem of suitable feeds for lobsters is critical. Sanders has developed a viable feed which is cost effective, nutritious, has readily available ingredients, is efficiently deliverable to the lobster in a culture system, and requires minimal amount of energy for manufacture and storage.

Sanders engineering of an effective grow-out system was perhaps the single most important challenge. Lobsters are both aggressive and territorial, and although their space, water quality and temperature needs are well known, growing lobsters to market size tends to be both capital and labor intensive.

The development of a viable culture system requires an economic data base upon which production costs resulting from each engineering improvement can be evaluated, and by which the economics of each element of the total system can be judged. Sanders venture team economic analysists have concentrated their efforts on the higher cost areas to achieve the required overall reduction in production costs. This effort has also provided potential investors with an objective evaluation of profit potential.

Disease is an ever present threat, particularly in intensive culture in water of elevated temperature. Sanders' management of Pediococcus (Golfkemia), Leucothrix (shell disease) and fungal infections is being accomplished by sound engineering design, prophylactic procedures and innoculation programs.

Characteristics sought by Sanders for a "domesticated" lobster include increased growth rate, higher percentage of meat, a strain more adapted to the culture environment, and an animal of a more docile temperament.

Lobsters require very high quality water compared with most species being cultivated. Additionally, in closed recirculating systems, ammonia, dissolved oxygen, pH level, and salinity must all be closely monitored and carefully manipulated.

Sanders aquaculture research and development efforts which started in January 1975 are ongoing in all of the areas mentioned. Each utilizes energy to a greater or lesser degree and is affected by the cost of energy at the chosen plant site as a design determinant.

A.2.4 Pilot Plant

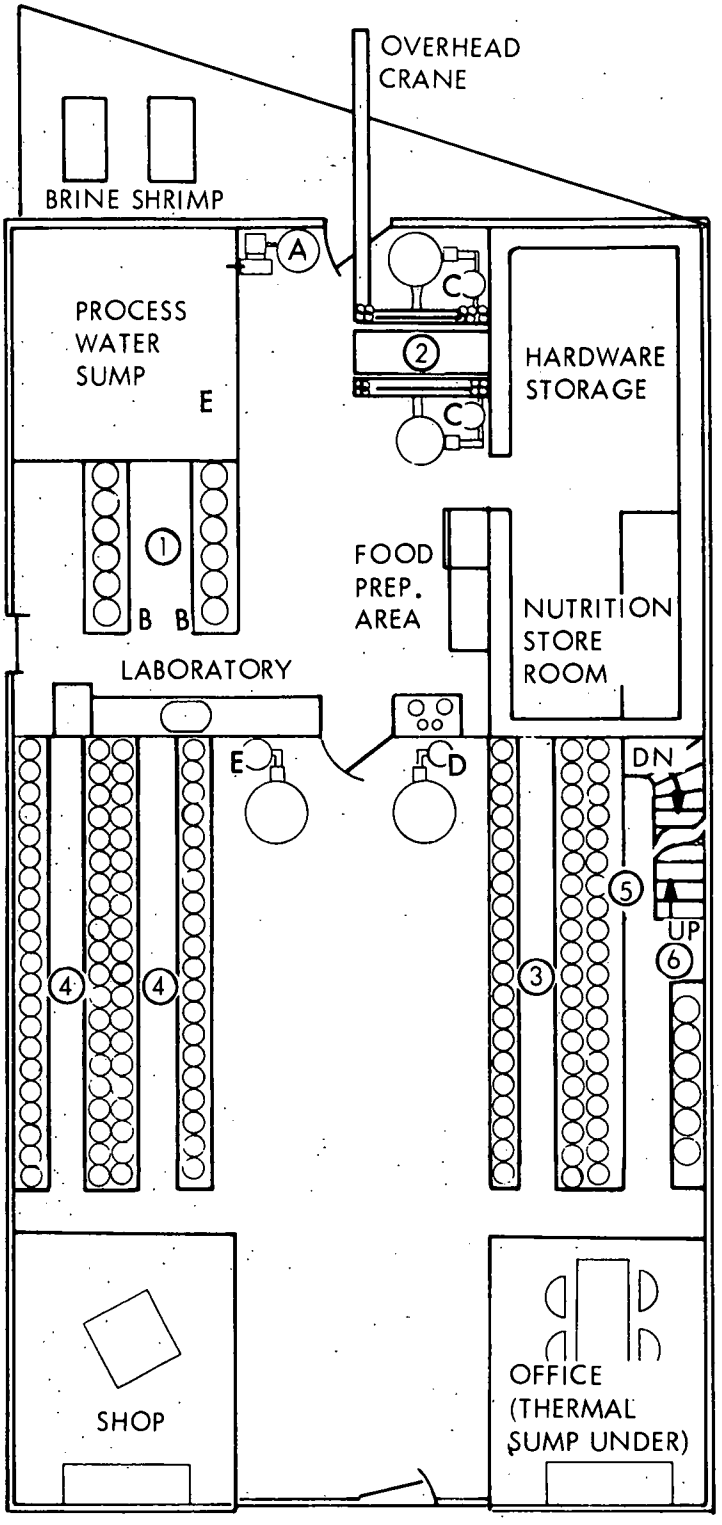
Based on research and development efforts over 15 years, a process has been developed for raising lobsters in captivity with sufficiently high yields and accelerated growth to make it a promising area of aquaculture. Now that the project has reached a pilot plant stage, and the major process requirements have been substantially determined, a further analysis of the economics of the process is in order, with a view towards ultimate full scale production.

The major requirements of the aquaculture process involve feeding, water change rate, and temperature maintenance. These requirements are common to aquaculture systems for lobsters, and other organisms requiring more or less individual attention inside a plant with a controlled environment.

The Kittery Pilot Plant culture system is depicted in Figure A-1 and A-2. It should be noted that for the purpose of providing the necessary flexibility and control required with the addition of a solar system, four sump pump units have been added to the baseline flow-through configuration.

The flow-through system utilizes pump "A" which brings water from deep mid-channels of the Piscataqua River and distributes it to sub-systems 1-6 via a manifold arrangement with appropriate throttling valves.

The aquaculture pilot plant is designed to adequately provide all equipment and system design concepts, operating procedures and economic factors prior to the decision to proceed on a full scale production facility. A description of the production facility is given in the following paragraphs.



1 - LARVAL FACILITY 18000 CAPACITY
2 - POST LARVAL FACILITY 3800 CAPACITY
3 - JUVENILE FACILITY 2720 CAPACITY
4 - ADULT FACILITY 1920 CAPACITY
5 - JUVENILE/ADULT RESERVE; 480 - 1360 CAPACITY
6 - BREED STOCK FACILITY 120 CAPACITY

Figure A-1. Kittery, Maine Plant

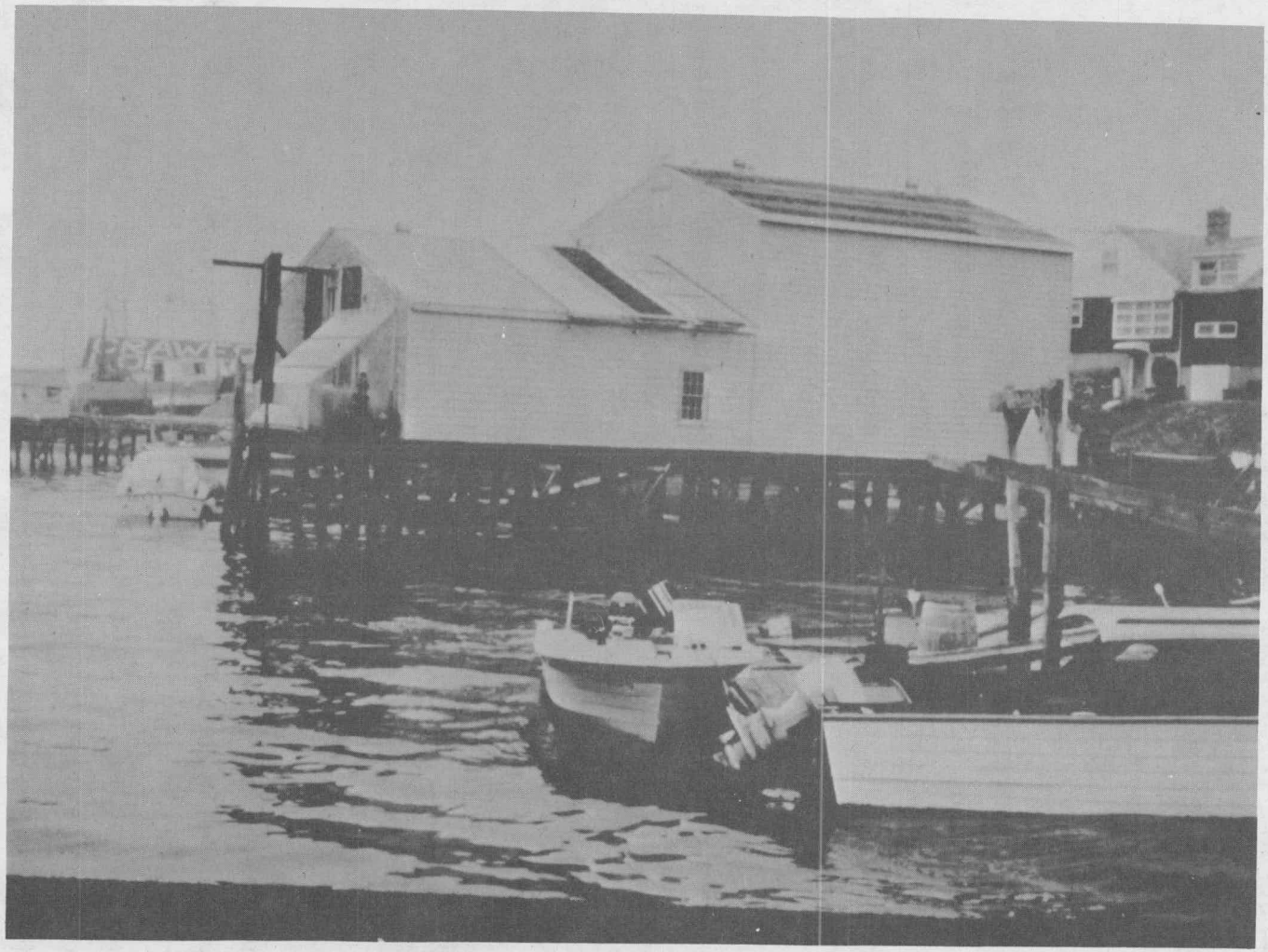


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Figure A-2. Sanders Lobsters Aquaculture Pilot Plant

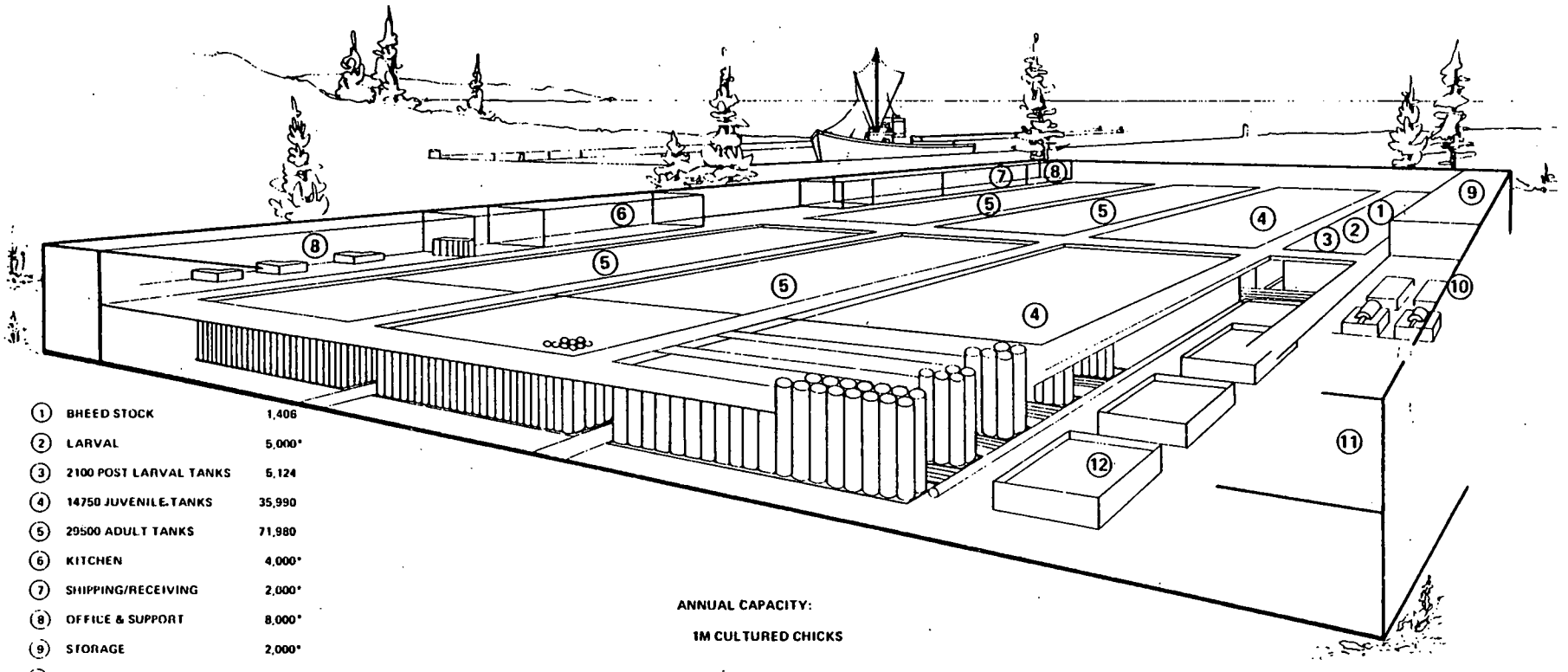
Scale-up problems have been inherent in the evolution of aquaculture programs from experimental to pilot plant and pilot plant to production stages. Sanders has avoided these problems by defining a modular system concept before the experimental program was implemented in 1974. This modular concept has been carried through all stages of development.

All tanks are arranged in double rows on a branched distribution manifold. A hydraulic module in the production plant consists of 1000 culture tanks, a recirculating pump, sump, filter and manifold distribution system. This approach provides the most economical and serviceable industrial environment for many types of culture.

The pilot plant will initially be equipped with a distribution system, pump and filter identical to that used in one production plant hydraulic module. However, until later in the development cycle, only one half of the module's tanks (500) will be provided. Actual operational flow rates, heat exchange rates, etc., will be duplicated and all parameters will be appropriately monitored with the same instrumentation to be used in the production facility.

A.2.5 A Commercial Aquaculture System

The commercial lobster aquaculture facility as developed by Sanders is shown in Figure A-3. The factors considered in its design in order of priority were: Individual rearing of animals from fifth stage (two weeks) to maturity (18-27 months) most effective use of building space, conservation of heat energy and minimal high grade energy consumption and appropriate human parameters. The Sanders patented cylindrical tanks offer features and flexibility not available with competing systems.



1	BREED STOCK	1,406
2	LARVAL	5,000*
3	2100 POST LARVAL TANKS	5,124
4	14750 JUVENILE TANKS	35,990
5	29500 ADULT TANKS	71,980
6	KITCHEN	4,000*
7	SHIPPING/RECEIVING	2,000*
8	OFFICE & SUPPORT	8,000*
9	STORAGE	2,000*
10	PUMP SYSTEM	3,000*
11	EMERGENCY EQUIPMENT	3,000*
12	BRINE SHRIMP	4,000*
	TOTAL SQ. FT.	130,000 SQ. FT.

*HALF HEIGHT

Figure A-3. Commercial Lobster Aquaculture Facility.

The 130,000 square foot facility provides a clear inside height of 30 feet for materials handling considerations in the culture areas. All other spaces are two levels with 12 foot ceiling heights providing ample support space.

Figure A-3 clearly delineates the area allocated to each function and culture stage in the facility. The water capacities of each system, pumping rates, heat loss and heat transfer characteristics of each area are tabulated.

Referring to Figure A-3, breed stock (1) are mated and brought to the point that the eggs are ready for hatching in 12-18 months. Elevated temperatures speed up the cycle and cooler temperature retard it enabling young to be hatched any time of the year. In practice, young are continually introduced to the system and adults are continuously harvested. Approximately 100 breed stock service the nearly 1 million lobster/year production.

Thirty larval systems (2) are busy year-round nurturing the newly born animals through the five molt larval stages in about two weeks. At this point, the animals are introduced to the post-larval system (3) where they spend the first third of their life before market size.

There are two more changes in habitat, one to the juvenile system (2) and the other to the adult system (5) before marketing.

The brine shrimp area (12) supplies the Artemia culture required for feeding both larval and early post larval lobsters. In the kitchen area (6) specially compounded diets are prepared for delivery to the animals.

APPENDIX B
CLIMATOLOGY AND INSOLATION AT PORTSMOUTH, NH

APPENDIX B
CLIMATOLOGY AND INSOLATION AT PORTSMOUTH, NH

B.1 INTRODUCTION

Considerable climatological data is available for Portsmouth, NH and its surroundings. Though this data is not slanted toward the quantities of interest for solar energy, it is adequate when combined with current, proven statistical and theoretical analyses to predict insolation and heat losses. In this field, monthly averages can be expected to fall within 10% of historical averages, although substantially larger fluctuations can be seen in shorter (weekly) periods. While considerable work is being done elsewhere* to model daily fluctuations of insolation, such data is much less important for an aquaculture application, where the time constant of the water in the plant is on the order of 20 days, than it is, for example, in-house heating or power plant operations where the relative cost of several days energy storage sets practical limits on the available averaging time for insolation and energy requirements.

Since these findings include a substantial need in the present aquaculture system for mechanical energy as well as for simple heating of the water, the insolation fluctuations will have some importance. As a result, a fossil fuel backup energy source will be provided for any energy demands beyond the selected energy storage capacity. This will make it possible to operate the plant in any contingency of weather conditions, with only minor variations in the cost of operation due to differences between actual weather and available model projections. Such a provision is especially appropriate to the New England climate, where intervals of adverse weather exceeding five successive days, though not common, have a significant probability and must be considered in any plant design.

*Reference to data base project.

Table B-1 contains excerpts of a climatological report on Portsmouth, NH, with typically available data on the 1954 - 1967 period.* Only more recent data, when it can be found+, includes more detailed insolation data. Table B-2 contains seawater temperature data for the same location for the period 1944 - 1958.

B.2 INSOLATION OF 60 DEGREES SLOPED ROOF AND ON TRACKING MIRRORS AT PORTSMOUTH NH

B.2.1 General

A tabulation is provided of insolation values appropriate to aquaculture or other flat plate solar collector applications or tracking mirror applications in the area. Following the tabulation of monthly and yearly averages, a detailed discussion is provided of the nature of the calculations and the assumptions on which these values are based. Some interpretation is given of columns other than the last, though they are not strictly essential for the above purposes.

The following definitions apply:

- N: maximum daily hours of sunlight
- n: equivalent daily hours at peak insolation
- H_o: hourly extraterrestrial insolation; i.e., without allowance for atmosphere, averaged over 24 hour
- H_{AV}: average horizontal insolation, (Btu/ft^2-hr)
- $H_{AV,T}$: average insolation, (Btu/ft^2-hr) on a sloped 60° slope surface, facing south
- R_{TOT}: multiplying factor (gain) to convert H_{AV} to $H_{AV,T}$ on a sloped surface

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*Cite the report heading

+Cite same data base project referred to in previous footnote.

TABLE B-1

SOURCE: NEW ENGLAND CLIMATIC SERVICE

READING, MASS. 01867

	TEMPERATURE (°F)												AVERAGE	
	MEANS			EXTREMES				MEAN DEGREE DAYS**	SKY				SUNSHINE	
MONTH	DAILY MAX.	DAILY MIN.	MONTHLY	RECORD HIGHEST	YEAR	RECORD LOWEST	YEAR		CLEAR	PRTLY CLDY	CLDY	CLD COV	% OF POSS	HOURS
(a)	13	13	13	13		13		13						
JAN	32.4	13.3	22.9	58	1966	-23	1957	1299	41	12	47	55	55	161
FEB	34.5	13.3	23.9	64	1957	-15	1962+	1152	39	14	47	56	60	177
MAR	41.8	22.3	32.1	76	1962	- 8	1967	1014	32	15	53	62	58	215
APR	53.3	31.7	42.5	92	1962	10	1967	668	30	14	56	65	55	221
MAY	66.0	40.9	53.5	94	1964	22	1967	364	33	17	50	61	60	273
JUN	75.4	50.7	63.1	96	1956	35	1967+	114	28	20	52	64	62	285
JUL	80.1	56.1	68.1	99	1963	40	1956	25	31	23	46	60	66	307
AUG	78.1	54.0	66.1	98	1955	33	1965	55	38	22	40	54	66	284
SEP	70.3	46.9	58.6	92	1965+	26	1962	217	38	18	44	56	63	236
OCT	61.0	37.3	49.2	88	1963	14	1966	488	42	15	43	53	60	205
NOV	49.1	29.5	39.3	76	1959	11	1957+	767	33	17	50	61	50	146
DEC	34.8	16.8	25.8	60	1966+	-12	1962	1207	38	15	47	57	55	155
YEAR	56.4	34.4	45.4	99	JULY 1963	-23	JAN 1957	7370	35	17	48	59	60	2665

(a) Average length of record, years.

\+ Also on earlier dates, months, or years.

** Base 65°F.

TABLE B-2

SURFACE WATER TEMPERATURE PORTSMOUTH, N.H.

YEARS	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	MINS	MAX	MIN
1944-48	34.1	32.9	36.1	41.7	48.0	54.0	59.1	61.7	59.4	52.4	46.9	39.2	47.1	68.	29.
1949-53	37.4	35.5	36.8	43.2	49.4	55.5	60.4	51.8	58.0	52.7	46.9	41.5	48.3	68.	30.
1954	36.0	34.8	38.0	43.5	49.7	56.0	58.8	58.3	57.5	53.4	47.4	41.3	47.9	66.	32.
1955	36.5	34.6	36.5	43.1	48.7	56.3	60.6	64.6	55.5	53.0	46.7	38.6	47.9	68.	32.
1956	36.5	35.6	35.3	39.6	46.0	55.7	58.7	60.2	58.4	52.2	47.9	39.8	47.2	64.	34.
1957	34.5	36.4	38.1	42.6	51.0	57.0	58.6	60.8	57.6	52.3	48.8	41.6	48.3	65.	32.
1958	37.3	35.3	38.4	43.5	49.7	53.0	57.6	57.7	56.6	51.8	47.7	39.4	47.3	63.	32.

SOURCE: New England Climatic Service

B.2.2 Results

A study has been made of insolation on level surfaces, surfaces sloping 60 degrees, oriented southward, and on "sun-tracking surfaces" for the Portsmouth, NH region. The 60 degree slope has been selected as one that favors winter energy production (as needed for aquaculture or building heat) at the expense of summer collection, as the best orientation for a stationary surface.

The horizontal surface figures are based on a statistical model in Duffie[1], and have been found to correlate reasonably well with a climatologist's data for Portsmouth. The correction for a stationary slope surface combines three elements:

(a) Beam Gain: Strictly dependent on relative orientation of the sun's rays and the surface, this gain was calculated at hourly intervals on a sampling day in the middle of each month. An average gain for the day was computed with a weighting proportional to hourly insolation applied to each hourly gain. Such an average was applied to each month, and for a yearly average, a weighting proportional to the number of days in each month was applied. The hourly and monthly samplings were judged consistent with illustrative examples in Duffie, and are considered commensurate with the validity of the statistical model projection and with the validity of future projections based on past data. In practice, the tabulated results apply as well to surfaces with at least a ±10 degrees misorientation in azimuth or elevation. Any radically different surface orientation will require repetition of the hourly insolation and gain computations; however, if the latitude is similar, the same insolation data may be re-used.

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(1) Duffie and Beckman pp 25-29, Solar Energy Thermal Processes, Wiley-interscience (1974)

(2) B. Liu and R. Jordan, Solar Energy 4 No. 3 (1960) The Interrelationship and Characteristic Distribution of Direct, Diffuse, and Total Solar Radiation

(b) Diffuse Gain: This figure depends only mildly on surface orientation, but very substantially on the fraction of the insolation that is diffuse. Since the diffuse gain was never more than half of the beam gain, the statistical model and historical model did not differ by more than 10%, the calculations were judged to be as good as any available projection.

(c) Reflective Gain: An accounting was made of the diffuse reflection contribution to collector insolation. To keep it conservative, a diffuse reflectivity ($\rho = 0.2$) was assumed for all months except December, January, and February, where $\rho = 0.7$ (for snow) was used. With the 60 degree roof slope, this gain never exceeded 10% of the overall figure, so that its accuracy has, at worst, a minor impact on the results.

The overall gain, a sum of the above three terms, is listed in Table B-3 and is used as a multiplier to convert the level surface insolation to a tilted surface value. Note that the level and tilted surface values for each month have very different annual patterns, with the sloped surface roughly doubling the available winter/summer performance: in fact, smoothing the year's variation to 1.59:1 in place of 5.23:1 for a level surface. While the total gain for the year totals 0.85; i.e., a 15% loss, the improvement of winter performance is of substantial value for many applications, including aquaculture among other applications.

For a tracking surface, the figures produced require qualifications as itemized below. They are based on the assumption that the surface is always normal to the sun's rays. This is reasonable for a one-piece disk in combination with a receiver. However, for multi-faceted mirrors, directing radiation at a single receiver, additional corrections are required (all of which involve more or less a reduction from the single mirror figures). Because one facet may shadow another (this

TABLE B-3

INSOLATION DATA FOR PORTSMOUTH, NEW HAMPSHIRE

Month	# Days	Daylight Hours N	Equivalent Hours at Peak Insolation n	Extraterrestrial H_O ** (Btu/h-ft^2)	Level Surface H_{AV} ** (Btu/h-ft^2)	Gain for 60° Slope R_{TOT}	Climatological Sunshine Hours/ Month	Gain for Single Mirror Tracker, Less Diffuse Radiation Fraction	60° Tilted $H_{AV,T}$** (Btu/h-ft^2)	Tracker H_{AV}, Direct Only (Btu/h-ft^2)
JAN	31	9.132	2.635	48.58	17.70	1.73*	161	2.29	30.67*	40.53
FEB	28	10.243	3.795	69.37	28.11	1.45*	177	1.92	40.7*	53.97
MAR	31	11.461	5.315	95.9	43.34	.99	215	1.49	42.90	64.58
APR	30	13.181	7.034	124.7	60.71	.737	221	1.23	44.74	74.67
MAY	31	14.332	8.268	144.4	73.42	.592	273	1.23	43.46	90.31
JUN	30	15.105	8.801	152.3	77.88	.539	285	1.24	42.0	96.57
JUL	31	14.491	8.501	147.0	75.46	.57	307	1.26	43.0	95.08
AUG	31	13.394	7.407	129.2	64.15	.693	284	1.26	44.46	80.83
SEP	30	12.092	5.76	102.0	46.73	.914	236	1.53	42.71	71.50
OCT	31	10.403	4.081	73.5	30.59	1.263	205	1.49	38.64	45.58
NOV	30	9.234	2.791	51.0	18.93	1.49	146	2.18	28.2	41.27
DEC	31	8.490	2.291	42.2	14.98	1.93*	155	1.51	28.91*	22.62
Yearly Average		11.803	5.565	98.48	46.09	.85		1.55	39.18	64.81

* Reflectivity = .7 Due to snow cover; elsewhere reflectivity assumed = .2

** Values given as hourly averages are daily total ÷ 24

varies with time of day as well as season) a tracking efficiency should be integrated over days and dates to yield a net correction to the single mirror insolation values. This tracking efficiency is, in turn, related to the placement of individual mirror facets. Though the efficiency is a function of the detailed placement of each mirror, the dependence is frequently summarized by a packing factor, which is the fraction of nominal surface area occupied by mirror facet area. In general, packing factor reduction increases average tracking efficiency. A tradeoff based on economic and system design factors generally can be performed to set an optimal combination of these two factors. Specifically, where substantial costs are involved, a detailed simulation analysis is appropriate, where a number of mirror placements are studied as a function of hour and date over a representative season. For estimating purposes, however, a reasonable "rule of thumb", based on examination of several cases for which the detailed analysis was performed, can be reasonably informative. The tracking efficiency will inevitably be below 100%, and the optimal packing factor is likely to be between 40% and 80%.

Another allowance that should be made for a focusing solar collector system is that it cannot function with diffuse radiation. Thus, when the net gain over horizontal surface radiation is calculated, the diffuse fraction (f_o) of this radiation must first be removed. Thus, the net gain over averaged horizontal surface insolation is essentially

$$R_{Tracker,\ AV} = (1-f_o)\ \frac{N}{n},$$

which is the "Gain for Single Mirror Tracker" column in Table B-3. Applying the figures in this column to the horizontal surface insolation, yields the insolation values to be used with a single mirror tracker. A multi-faceted mirror will entail a reduced (from 100%)

tracking efficiency, which will also be a function of the packing fraction. Since all these figures are based on daily and monthly averages, specific designs must allow for peak/average ratios, and for whatever thermal load leveling system (e.g., energy storage) is part of the energy utilizing system.

Figure B-1 shows a 1.55 average yearly gain for a multiple mirror tracking system over a stationary horizontal surface, which is of value in helping offset the expense of the tracking system. To derive this figure, enough allowance was made for diffuse radiation (assumed usable by the horizontal surface collector, but not by the tracking and focusing system) that the gain would have been 2.0, where the sky is always clear. This figure is stated to illustrate the advantage of a desert climate (provided it fits the nature of the application, and has equally favorable economics).

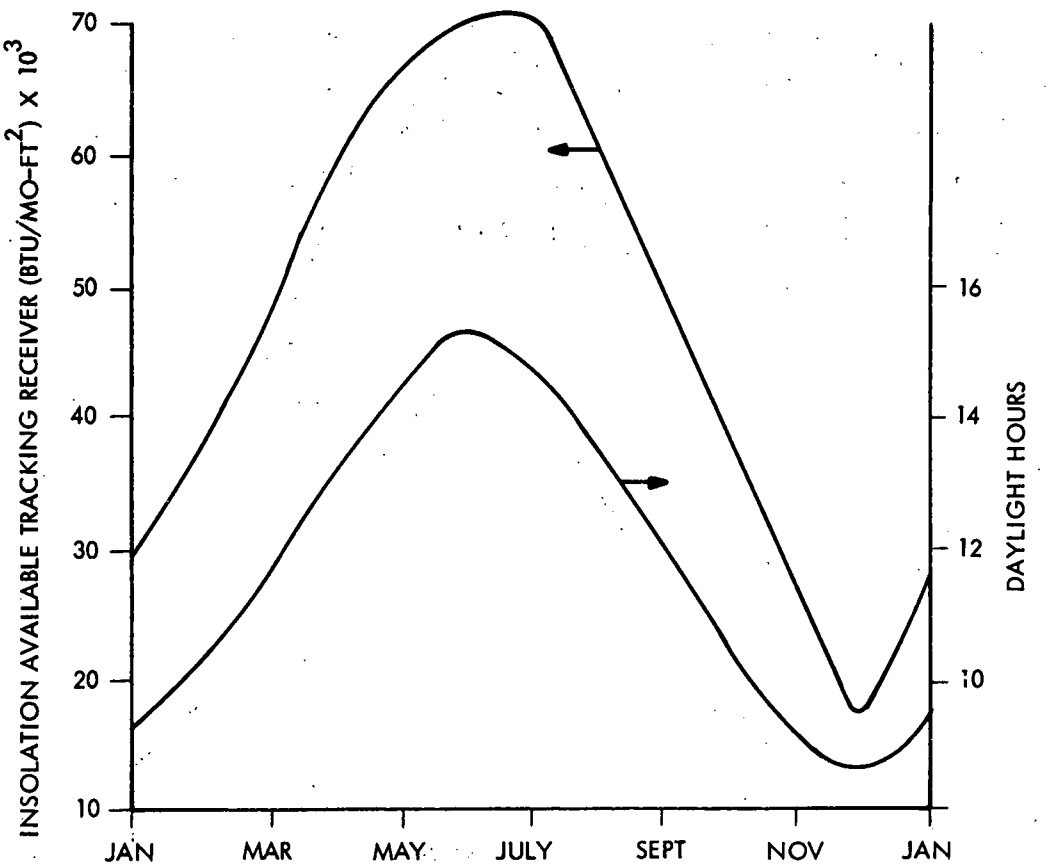

Figure B-1. Monthly Insolation for Tracking Receiver Portsmouth, N.H.

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