DEVELOPMENT AND APPLICATION OF DECENTRALIZED ENERGY SYSTEM UTILIZING NON-CONVENTIONAL ENERGY SOURCES -SOLOJIPALLY PROJECT-

SOLAR THERMAL POWER PLANT SYSTEM ANALYSIS

JULY 15, 1981

BHEL PROJECT NO: DYN-80-240-83
JPL TASK PLAN NO: 70-980

IMPLEMENTING AGENCY

BHARAT HEAVY ELECTRICAL LIMITED, HYDERABAD, INDIA

COLLABORATING AGENCY:

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SOLAR THERMAL POWER PLANT

SYSTEM ANALYSIS

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July 15, 1981

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1.0 INTRODUCTION

The basic objective in the development of the solar thermal power generation system is to design, fabricate and install a system that would fulfill the essential energy needs of the selected community. The system design will, therefore, have to be based on available technologies in order to ensure reliability of the system. In addition, the system should be simple in operation and compatible with the environmental infrastructure.

A site, which has been selected for the installation and testing of the solar power plant, is a village, named Solojipally in Medak district of Andra Pradesh State. The village is situated about 105 km from the Corporate R&D Division, BHEL, Hyderabad. Some important details of the village are as follows:

Geographical Location	:	18° North Latitude
	:	78° East Longitude
Situation	:	Andhra Pradesh State 110 km from Hyderabad
Road Communication	:	Situated by the side of a metalized road connecting subdivisional headquarters
Status of Electrification	:	Unelectrified
Population	:	424
Number of Houses	:	71
Occupation	:	Farming and Labour
Total Land Under Cultivation	:	165 Hectares
Land Covered by Irrigation	:	60 Hectares
Land not covered by any Irrigation	:	105 Hectares
Mode of Irrigation:	:	Monsoon fed tanks under gravity flow and wells
Crops Grown	:	Rice, Sorgham
Domestic Lighting	:	Kerosene Oil Lamps
Cooking	:	Firewood
Drinking Water	:	Lifting manually from open wells
Water Table for Pumping	:	20-30 metre below surface depending on location

The village is situated in a semi-arid zone where the solar insolation is fairly good throughout the year, except the monsoon period of about three months. The annual average of direct solar radiation is about 500 watts/m^2. The wind regime at the selected site is rather low, the annual average wind speed being 10 kmph. There is a possibiltiy of utilization of bio-mass energy available in and around the community. The quantum of biomass available within the village is 850 kg/day.

The community is predominantly agrarian and a large percentage of the cultivatable land depends on monsoon rain. The energy requirement, therefore, is primarily for irrigation which can bring the greatest benefit to the community through an increase in agricultural productivity.

Due to bugetary limitations, the output from the solar thermal system has been limited to the order of 12 kWe. It is recognized that the capacity of the envisaged system will not be able to fulfill the total energy requirement of the community.

The following pages contain the brief details of the overall system design for a solar thermal power plant capable of delivering up to 20 kWe power depending on the solar insolation value. The selection of thermodynamic cycle and subsystems/components have been made keeping in view of the available technologies which are suitable for rural application. The power plant uses point focusing collectors and the steam Rankine cycle. The power conversion system is of the central conversion type with the performance depending on the availability of a suitable steam engine. No storage facilities have been provided for the operation of power plant during non-sunshine hours. However, provision has not been made for incorporating a suitable buffer storage which can take care of transient fluctuation in solar energy input. The report is also intended to provide sufficient inputs for detailed design of subsystems/components required for the power plant.

2.0 SELECTION OF A THERMODYNAMIC CYCLE

Out of two major classes of thermal energy collection and power generation systems, namely centralized receiver and distributed receiver, the latter system has been opted, as it is suitable for small output capacities envisaged in this project. A power conversion system, associated with distributed receivers, may be based on:

- Brayton cycle,
- Stirling cycle, and
- Rankine cycle.

Brayton Cycle:

The Brayton cycle, which essentially demands employment of an appropriate gas turbine for conversion of thermal energy into mechanical energy, requires high temperature potential (600-750°C). This system would require the availability of high temperature concentrating collectors and high temperature gas technology. The design methodology and technology for manufacture of such collectors and fabrication of gas turbines are still not established in India. The possibility of employing a Brayton system, under this project, have seriously been looked into during discussions with JPL team. At JPL also, the first version of Brayton system with appropriate receiver has not yet been tested. Considering these factors, this cycle has not been adopted.

Stirling Cycle:

The power conversion system based on Stirling cycle has high promise because of their high efficiency, minimum auxiliary power consumption and possibiltiy of using such engines in a distributed way. However, the approach to design and fabrication technology of Stirling engines are not well understood in India. Moreover, the temperature of operation of hot air cycle is in the order of 600 to 800°C, posing problems on the availability of suitable concentrators and receivers using high temperature materials in India. Although JPL is working with the development of a power conversion system based on this cycle, the first version of this receiver will not be tested until the month of July '81. Based on the present development and experience at JPL, it is felt that, this is a new technology, and requires thorough performance and reliability testing before it can be put into use. Therefore, it is too early and risky to adopt such a system.

Rankine Cycle:

The Rankine cycle operation is limited to the lower temperature range of up to about 600°C and it has lower efficiency than both Brayton and Stirling cycle. However, this system is well proven and many components required for this system can be fabricated in India. Therefore, a power generating system based on the Rankine cycle has been chosen for this project because of its operational reliability and simplicity.

3.0 SELECTION OF RANKINE CYCLE SUBSYSTEMS

A power conversion system, based on the Rankine cycle with distributed receivers, will operate on the central or distributed approach. In the central conversion, thermal energy from the receivers is transported to a central location, where it is converted to electricity using a single heat engine/generator assembly. In the distributed power conversion, by constrast, each collector will have a small heat engine/generator assembly suitable to the capacity of the collector which forms, preferably, a part of the receiver assembly. The distributed system may be more economical as there is the possibility of cost reduction due to mass production of identical units and is more efficient as piping network for the thermal transport system is avoided saving high thermal losses compared to electrical transmission loss, provided there is no substantial difference in the efficiencies between small distributed prime movers and the larger central one. However, the distributed approach may increase maintainance activities and complexity in control system.

For this project both of the systems have been considered. However, for a solar power plant of capacity in the order of 15 to 20 kWe, the central approach may be more feasible as very small capacity power conversion systems would have to be incorporated in the distributed approach.

The subsystems of a solar thermal power plant, based on the above concept, are:

- Energy Collection Subsystem
- Energy Storage Subsystem
- Energy Conversion Subsystem

3.1 ENERGY COLLECTION SUBSYSTEM

It is known that the Rankine cycle efficiency increases with an increase in the cycle operating temperature and a decrease in rejection temperature. For a power conversion system, incorporating a water cooled condenser, the rejection temperature is more or less fixed. Therefore, to increase operating temperature, the collector capable of providing a high temperature will be a better choice.

Therefore, point focusing collector with faceted mirrors as a reflective surface has been chosen. This collector assembly is expected to deliver thermal energy at high temperature more efficiently than other available solar collectors such as flat plate and line focusing. The collector is polar mounted which may have a less complicated structure than the azimuth mounted type and is also suitable for the latitude of the selected site. It is desirable to have a collector diameter in the range of 7 to 9 m, which is fairly large and may not pose much difficulty in fabrication and handling in India.

The receiver is of the cavity type with a circular tube fl channel and well insulated. The receiver and collector assembly is expected to raise the temperature of the fluid in the order of 400 to 600°C. The receiver heat transfer fluid is water which will be converted to superheated steam. The criteria for the selection of this fluid has been described in article 3.3.

3.2 ENERGY STORAGE SUBSYSTEM

Thermal energy storage can be accomplished through several routes, such as:

- Sensible heat storage
- Chemical reaction storage
- Phase change material storage

Sensible heat storage at temperature (400 to 500°C) requires a large volume of storage media and hence it is highly uneconomical. The solid (pebble) bed storage system with oil as the heat transfer medium are also uneconomical due to the requirement of large volumes of oil for the same storage capacity when compared with phase change material storage. Moreover, selection of suitable solids withstanding high temperatures and possessing favorable properties for thermal storage, has to be made.

Chemical reaction storage is comparatively a new concept in which energy is stored by the heat of chemical reaction. Suitable chemicals capable of delivering energy at required temperatures, and having adequate life cycles, are yet to be identified.

Phase change materials with solid to liquid transitions are quite useful due to relatively small accompanying volumes. Moreover, the energy is available at a constant temperature and the complications of maintaining stratification is avoided. However, several problems like minimization of charging and discharging timing with minimum temperature differences, between the heat transfer medium and the phase change material have to be resolved and practical evaluation of the life cycle has to be made, before using such a system on a large scale.

Keeping in view the present state of the art, both at BHEL and JPL, it is decided to have no thermal storage in the system to take care of diurnal variation of insolation and for the operation of the power plant during non-sunshine hours. However, buffer storage of suitable capacity will be incorporated in the system to take care of transient fluctuations in input energy.

3.3 ENERGY CONVERSION SUBSYSTEM

While the dimensions of the collector field and the capacity of the thermal storage will directly affect the cost of the power plant, the prime mover influences the overall efficiency and economy quite significantly. The basic criteria for the selection of a prime mover for the solar energy systems are:

- It should have highest possible efficiency to allow reduction of collector area as much as possible.
- Working fluid should be available, be stable at operating temperatures and preferably nontoxic.
- From the thermodynamic point of view, the saturated vapor pressure should be low to reduce the thermal inertia, and the slope of the saturated vapor line in the temperature-entropy diagram should be as steep as possible to avoid either a higher degree of condensation or superheating during expansion.

Working Fluid.

The suitable working fluids that are envisoned, for solar power plants based on the Rankine cycle, are steam, toluene, benezene and monochlorobenzene. However, the thermal stability of organic fluids like benzene and monochlorobenzene are questionable at the high operating temperatures. Toluene could be used, where the plant is to be designed for operation above the critical point for this fluid. Systems with toluene as a working fluid, are in the developmental stage. Therefore, the only working fluid which is suitable for the present system is steam.

Prime-Mover:

The prime movers, which are suitable for Rankine cycle power conversion systems, are steam engines, turbines and screw expanders. Out of these three, the screw expander, being most efficient, has the highest priority for the solar application. It has been confirmed during the first conference meeting at JPL, that the screw expanders of the required capacity are not available in the market. Steam turbines in the output range of 15-20 kWe are also not available in the market. Steam engines in this output range are available, and JPL is carrying out modifications and testing of steam engines for solar operation. These engines have considerably higher efficiency. For the present power plant, a suitable steam engine is to be used.

Total Energy Concept:

The system with partial expansion up to one ata and utilizing the rejected energy for the operation of a vapor absorption refrigeration system will have an overall system efficiency (of the order of 0.5 to 1%) higher than the systems with full expansion up to 0.1 ata. But in this project we would like to go for full expansion scheme because, there is no demand for cold storage or hot water in the village, selected as the experiment site.

Single and Double Loop Concept:

In a double loop system a heat transfer fluid, commonly an oil, receives heat in the receiver which is transferred to the working fluid in the boiler of the power conversion subsystem. Where as in single loop concept, the working fluid is heated directly in the receiver. Some major coments on both the concepts from the performance and operational considerations have been presented below:

- The selection of a suitable oil capable of working up to 500°C-600°C is questionable. Therefore one has to either decrease the working temperature level at the cost of performance or use molten salt with additional complications.

- Based on the present experience, the efficiency of an oil receiver is about 10% less than the receivers generating superheated steam.

- The oil system has to be free from air and oxygen to avoid faster decomposition of oil at higher temperature. In some cases, the viscosity of oil is high at atmospheric conditions and hence a preheating arrangement is required which is difficult to provide. Special welding is required to have leakproof system with oil at high temperature.

- The oil system requires an additional heat exchanger and one or two pumps, depending upon whether a thermal storage is incorporated or not. The auxiliary power consumption is high and this system is comparatively costly due to the high cost of oil.

- Thermal losses may remain almost the same in both cases, because the loss due to the larger size of the steam transport piping in the single loop system may be compensated with the loss due to high temperature oil transport before the receiver. But temperature drop in the oil transport system is much less than steam transport system. However, one has to account for the temperature gradient required between oil and steam in the heat exchange.

- Working pressure in the oil system is much less but flow rate is high. Hence pressure drop may be comparable in both the cases.

- The control scheme for the double loop system is comparatively simplier than for single loop system.

Based on the above considerations, single loop principle has been adopted for the present system.

Auxiliary Power:

It may be mentioned that in a power generation system, the auxiliary power requirement is of the order of 20-30%. Moreover, power is required for starting of the system. A bio-gas power generating system can be used to meet the requirement of auxiliary power to reduce the cost of the solar power plant. Such a bio-gas backup system will operate as a complementary system to the solar power plant during dark periods. The bio-gas plant of suitable capacity will be made available by BHEL.

4.0 ANALYSIS OF INSOLATION DATA

For the design of a cost effective solar power plant, insolation data will play a vital role in fixing the individual subsystem parameters. The solar insolation data at the project site are not available. However, the site is situated almost at the same latitude as Hyderabad (which is 17.8 degree north) and also possesses same climatological conditions that of Hyderabad throughout the year. Therefore, it is justifiable to use the insolation data obtained at Hyderabad for the analysis of the solar power plant system to be tested at the selected site.

Direct solar radiation on a horizontal surface has been measured and tabulated on an hourly basis for each day of the year. From these data, the values normal to the dish have been evaluated using the following expression:

Radiation normal to the dish = Radiation on the horizontal surface/$\text{Sin}A_h$

where, $\text{Sin}A_h$ = Cos B Cos C Cos D + Sin B Sin C

B = Latitude angle

= 17.8 degree for Hyderabad

C = Declenation

D = Hour angle

Table 1 gives the monthly average values of direct solar radiation on horizontal surfaces in Watts/m^2 for various hours (from 0800 hrs to 1700 hrs) of the day for the year 1980. The monthly average values of radiation normal to the dish for the same year have been given in Table

Table 1. DIRECT SOLAR RADIATION ON HORIZONTAL SURFACE

(Monthly Average Value, Watts/M^2)

Year 1980

MONTHS \ HOURS	8-9	9-10	10-11	11-12	12-13	13-14	14-15	15-16	16-17
January	172.0	362.7	497.3	578.6	595.7	541.7	444.2	295.5	132.4
February	180.8	374.3	495.1	595.6	525.6	567.5	445.2	306.3	173.5
March	213.6	373.2	505.3	608.7	586.0	491.9	406.7	296.1	157.9
April	241.8	382.3	492.0	639.9	582.9	520.9	375.9	272.1	129.9
May	315.2	493.5	615.5	685.4	671.3	594.7	458.7	288.2	147.6
June	126.8	242.1	296.4	288.8	302.7	281.1	210.3	169.5	111.3
July	114.7	184.9	221.1	230.0	199.2	180.3	180.7	123.2	111.2
August	115.1	162.3	175.8	232.2	230.1	240.6	264.5	192.7	114.2
September	202.5	291.9	363.6	431.8	452.2	377.1	322.8	216.4	126.1
October	361.9	546.8	688.5	715.9	705.4	625.5	515.1	319.0	137.8
November	231.9	396.0	519.0	560.5	563.2	494.7	384.9	246.5	90.7
December	196.8	341.8	449.0	539.1	549.8	481.3	398.7	257.0	121.2
YEARLY AVERAGE	206.1	346.0	443.2	508.9	497.0	449.8	367.3	248.5	129.5

2. These values include the days with very poor insolations also. Hence, to have a more realistic analysis of insolation data, applicable for solar energy systems, the days with 0.5 kW/m^2 direct insolation for at least 5 hours duration are considered and monthly average of these values have been presented in Table 3. Figure 1 through 12 represents the monthly average value of direct solar radiation normal to the dish both for all the days and for those days with 0.5 kW/m^2 insolation for at least 5 hours duration. One can find from these figures and Table 3 that the duration of 0.5 kW/m^2 insolation level now ranges for 7 to 9 hours when presented on monthly average basis. It is because, the minimum 5 hours span for 0.5 kW/m^2 insolation level considered, is not confined to the same hours of the day. Under this circumstance it is not possible to choose insolation level from these monthly mean values for system analysis.

Table 4 gives the number of days in the year with a 0.5, 0.55, and 0.6 kW/m^2 direct solar radiation normal to the dish for at least 5 hours daily. It can be seen from the table that the plant can be operated for 207, 192 and 166 days in a year respectively for insolations 0.5, 0.55 and 0.6 kW/m^2. Keeping the operation of the plant to approximately 200 days in a year, 0.55 kW/m^2 insolation level has been chosen as the start point for the power plant to deliver an output. In the present system, this value will determine the starting load output from the plant.

It is also necessary to indicate the peak insolation level in order to determine the maximum output from the power plant. This peak should not be the isolated peak, but rather one which is attained in a considerable number of days. Table 5 gives the number of days with

Table 2. DIRECT SOLAR RADIATION NORMAL TO DISH
(Monthly Average Value, Watts/M2)

Year 1980

HOURS / MONTHS	8-9	9-10	10-11	11-12	12-13	13-14	14-15	15-16	16-17
January	496.2	684.2	742.5	764.5	759.4	720.7	672.3	570.9	399.5
February	456.8	638.3	673.3	719.8	728.3	685.7	605.0	520.9	439.2
March	437.2	548.6	610.9	665.3	623.6	546.5	509.3	461.7	358.5
April	421.8	504.7	550.9	658.6	589.9	554.3	452.3	407.0	280.0
May	519.7	631.3	676.2	696.9	673.7	627.2	544.9	420.8	302.4
June	212.5	314.1	329.3	295.8	304.6	259.3	246.6	240.8	216.1
July	198.4	244.0	247.7	236.1	199.7	188.0	209.3	172.3	211.6
August	203.8	215.9	197.6	238.7	231.8	253.1	312.0	278.6	232.3
September	367.6	397.5	419.5	458.3	474.8	420.5	412.4	352.2	310.2
October	704.8	759.2	848.8	818.2	805.5	769.2	744.1	613.4	449.0
November	521.9	648.1	711.1	707.3	709.2	672.5	620.8	541.4	358.7
December	459.6	582.0	642.7	715.0	733.2	700.8	700.8	635.5	594.1
YEARLY AVERAGE	416.7	514.0	554.2	581.2	569.5	533.2	502.5	434.6	346.0

Table 3. DIRECT SOLAR RADIATION NORMAL TO DISH

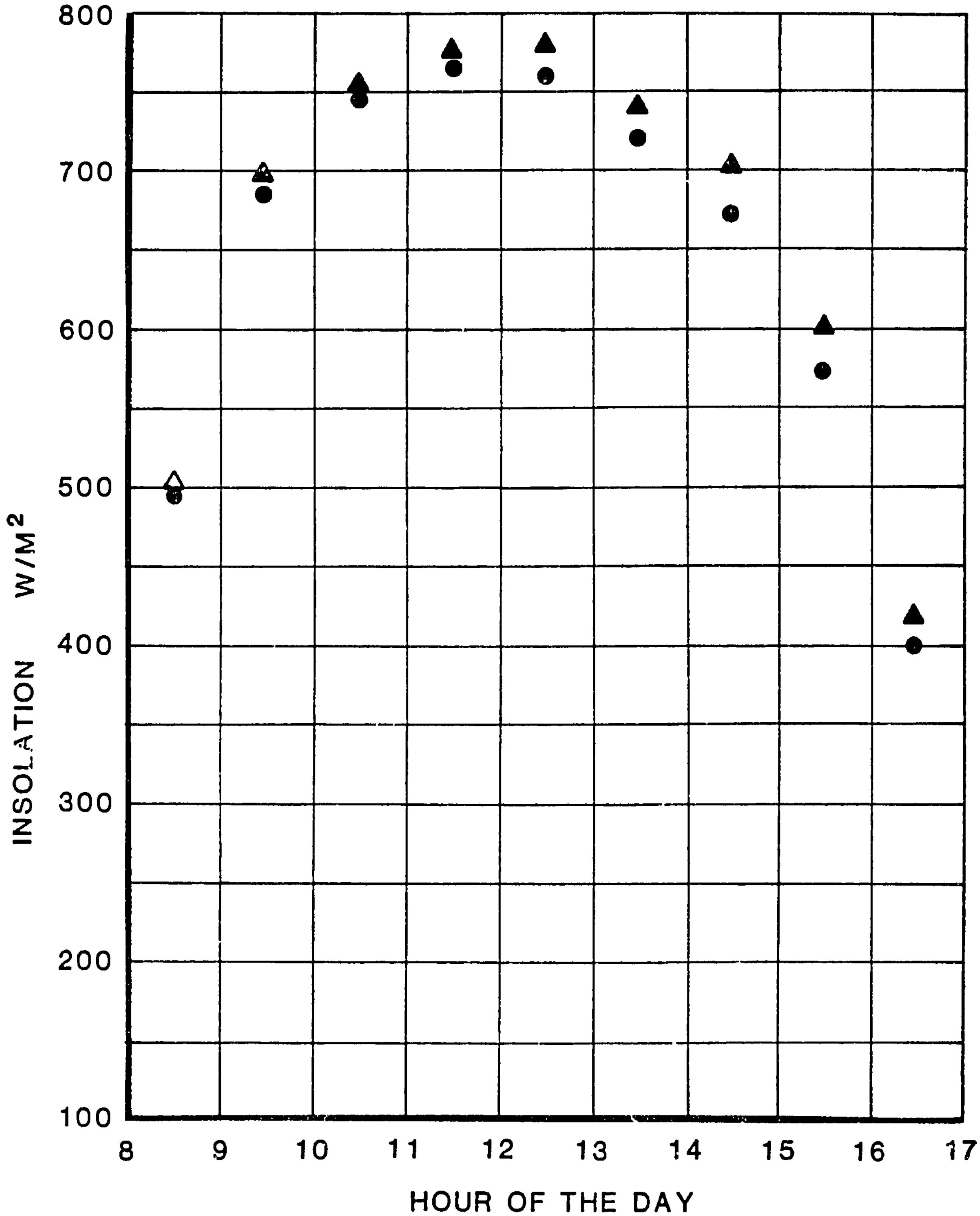
(Monthly Average Value with 0.5 kW/M^2 for at Least 5 Hrs. a Day)

Year 1980

HOURS / MONTHS	8-9	9-10	10-11	11-12	12-13	13-14	14-15	15-16	16-17
January	505.2	697.0	752.8	774.2	778.9	739.6	701.2	599.4	416.3
February	458.4	640.9	676.0	722.4	739.4	667.7	610.1	526.9	442.5
March	479.6	593.1	646.0	695.8	684.2	610.5	563.5	516.3	399.2
April	458.3	548.0	578.3	680.4	626.8	643.5	540.1	494.1	309.3
May	546.5	661.2	701.7	722.5	703.1	658.2	578.7	455.1	324.4
June	-	-	-	-	-	-	-	-	-
July	-	-	-	-	-	-	-	-	-
August	113.0	131.2	169.8	642.9	705.3	734.3	746.9	617.2	403.2
September	679.6	725.0	754.6	742.7	707.8	701.2	653.9	625.2	516.8
October	737.2	819.5	868.2	839.6	833.6	790.5	758.5	639.9	452.3
November	595.7	758.6	845.7	831.7	851.5	806.3	761.4	673.2	428.6
December	543.8	686.4	757.3	854.5	883.4	869.7	835.2	758.3	696.2
YEARLY AVERAGE	511.7	626.0	675.0	750.7	751.4	722.2	675.0	590.6	438.9

FIG. 1 DIRECT SOLAR INSOLATION NORMAL TO DISH
MONTHLY AVERAGE

JANUARY,80
HYDERABAD



● — FOR ALL DAYS

▲ — FOR DAYS WITH 0.5 KW/M^2 INSOLATION FOR MINIMUM 5 HRS

FIG. 2 DIRECT SOLAR INSOLATION NORMAL TO DISH
MONTHLY AVERAGE

FEBRUARY,80
HYDERABAD

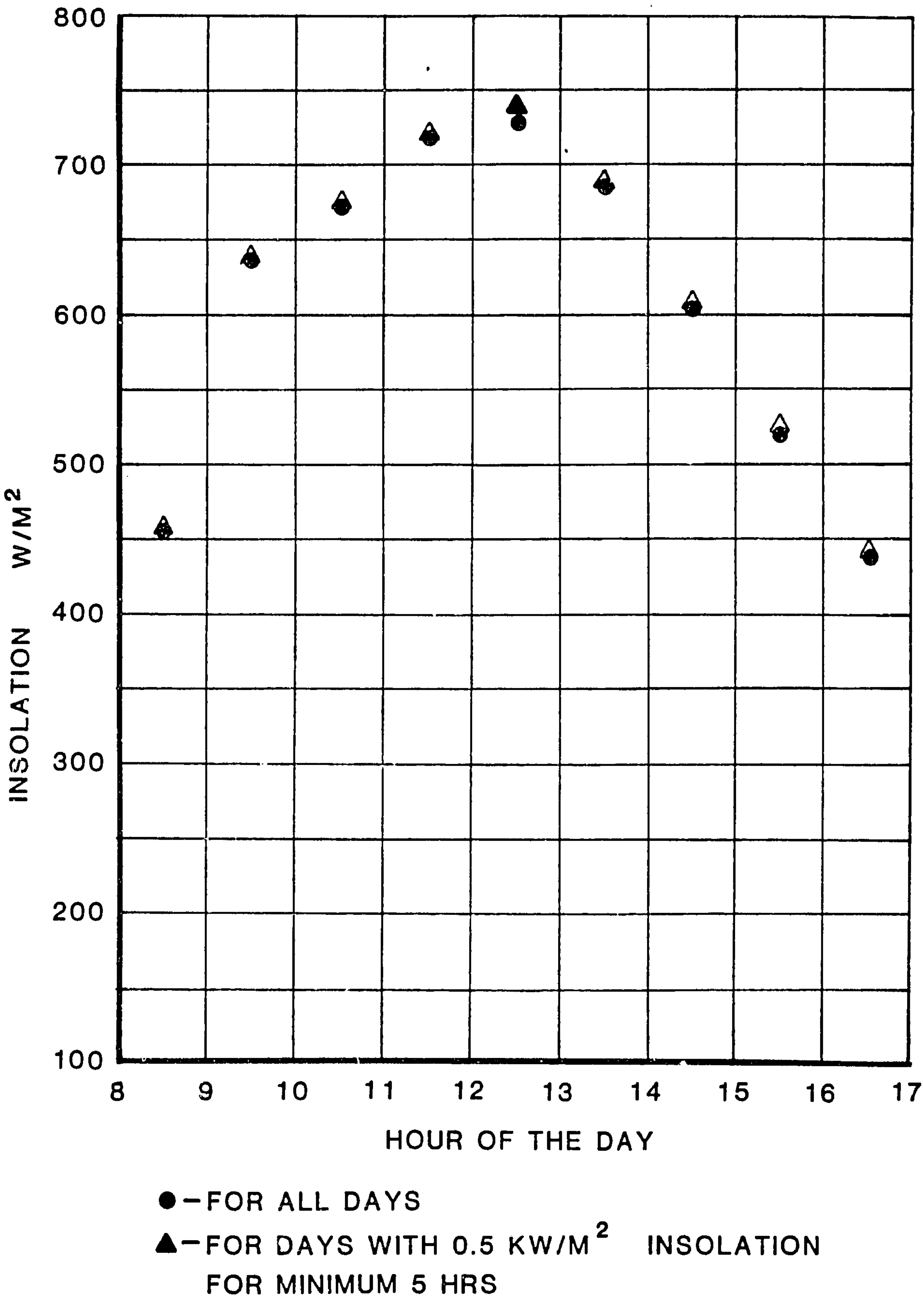


FIG. 3 DIRECT SOLAR INSOLATION NORMAL TO DISH
MONTHLY AVERAGE

MARCH,80
HYDERABAD

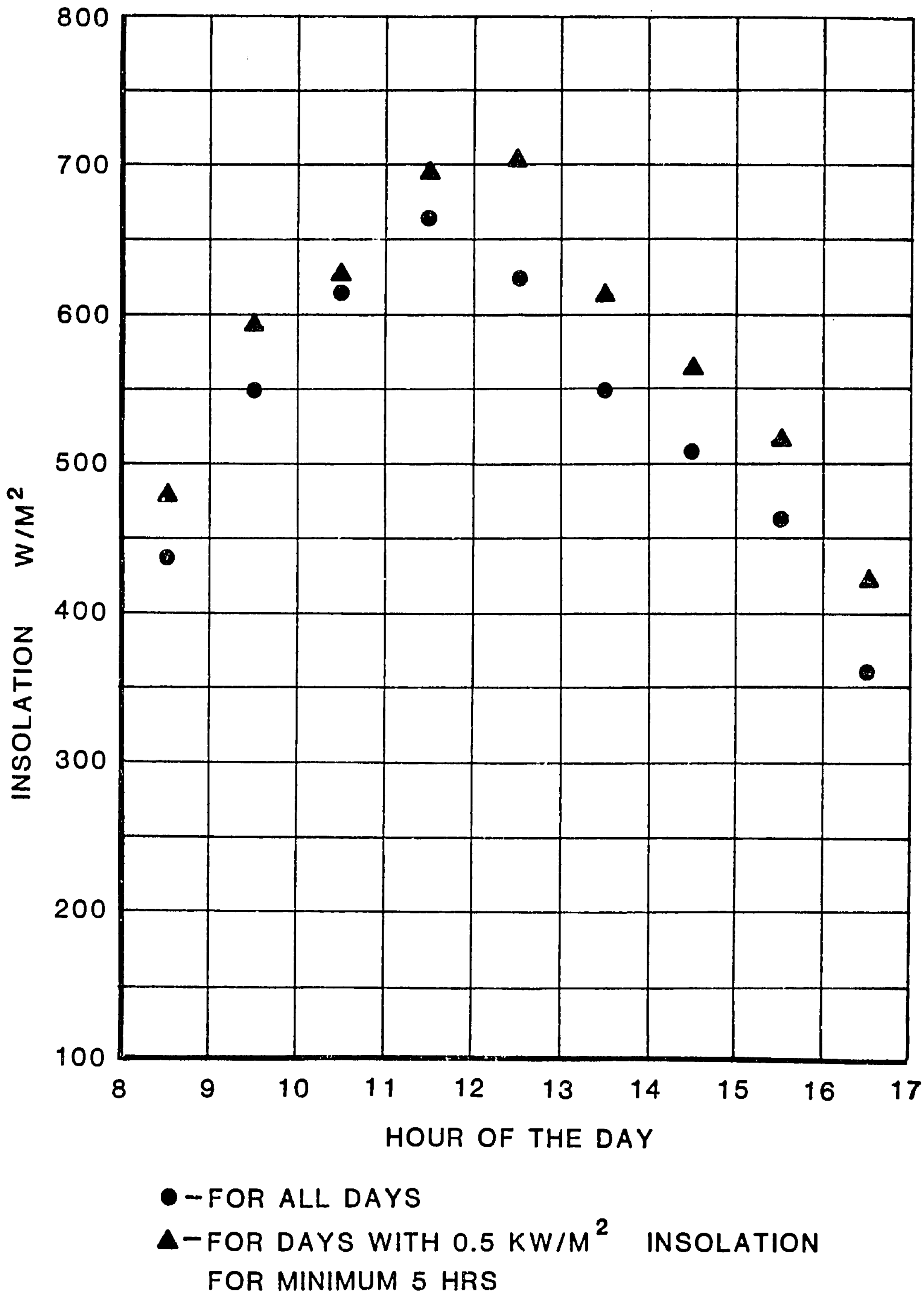
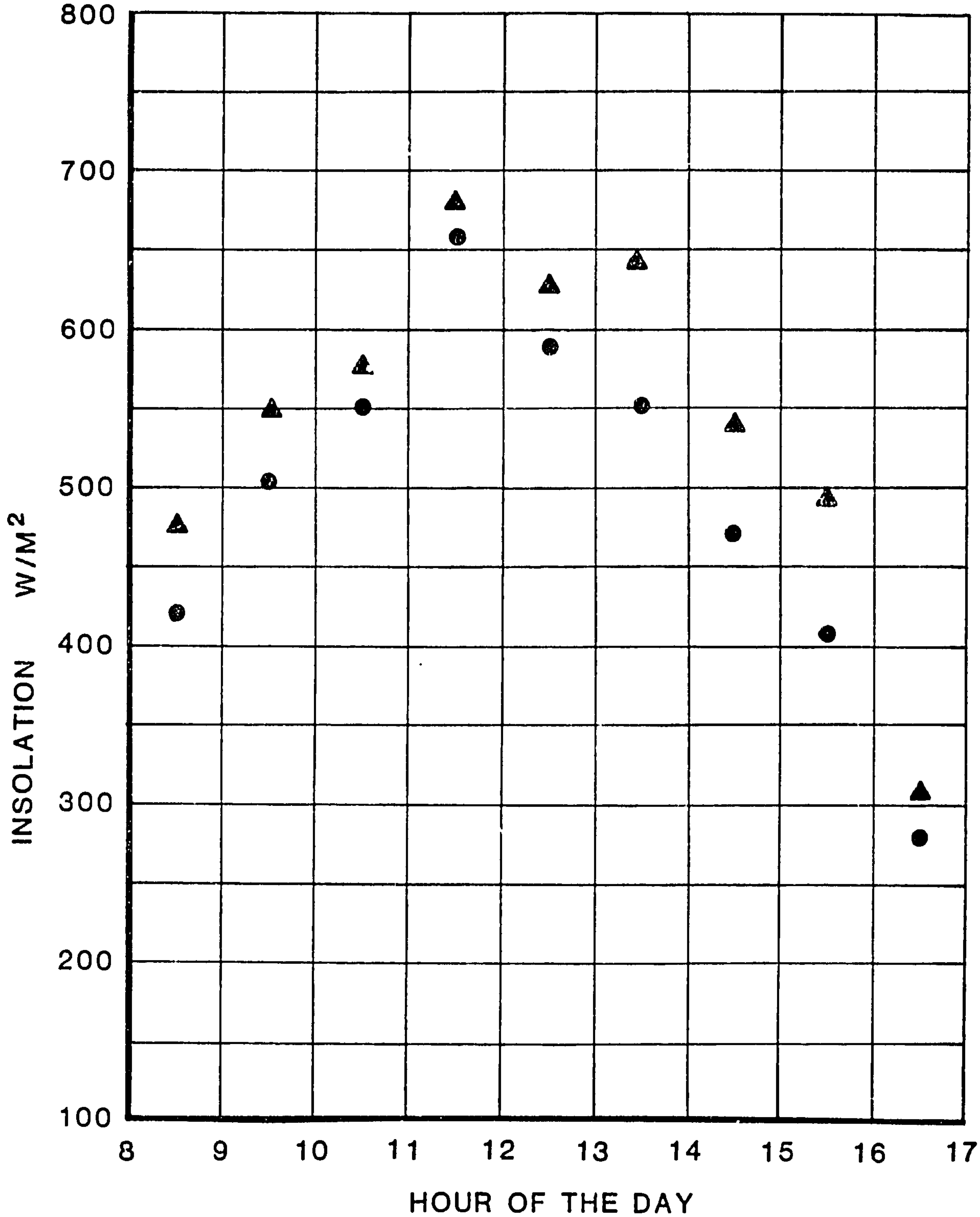


FIG. 4 DIRECT SOLAR INSOLATION NORMAL TO DISH
MONTHLY AVERAGE

APRIL,80
HYDERABAD

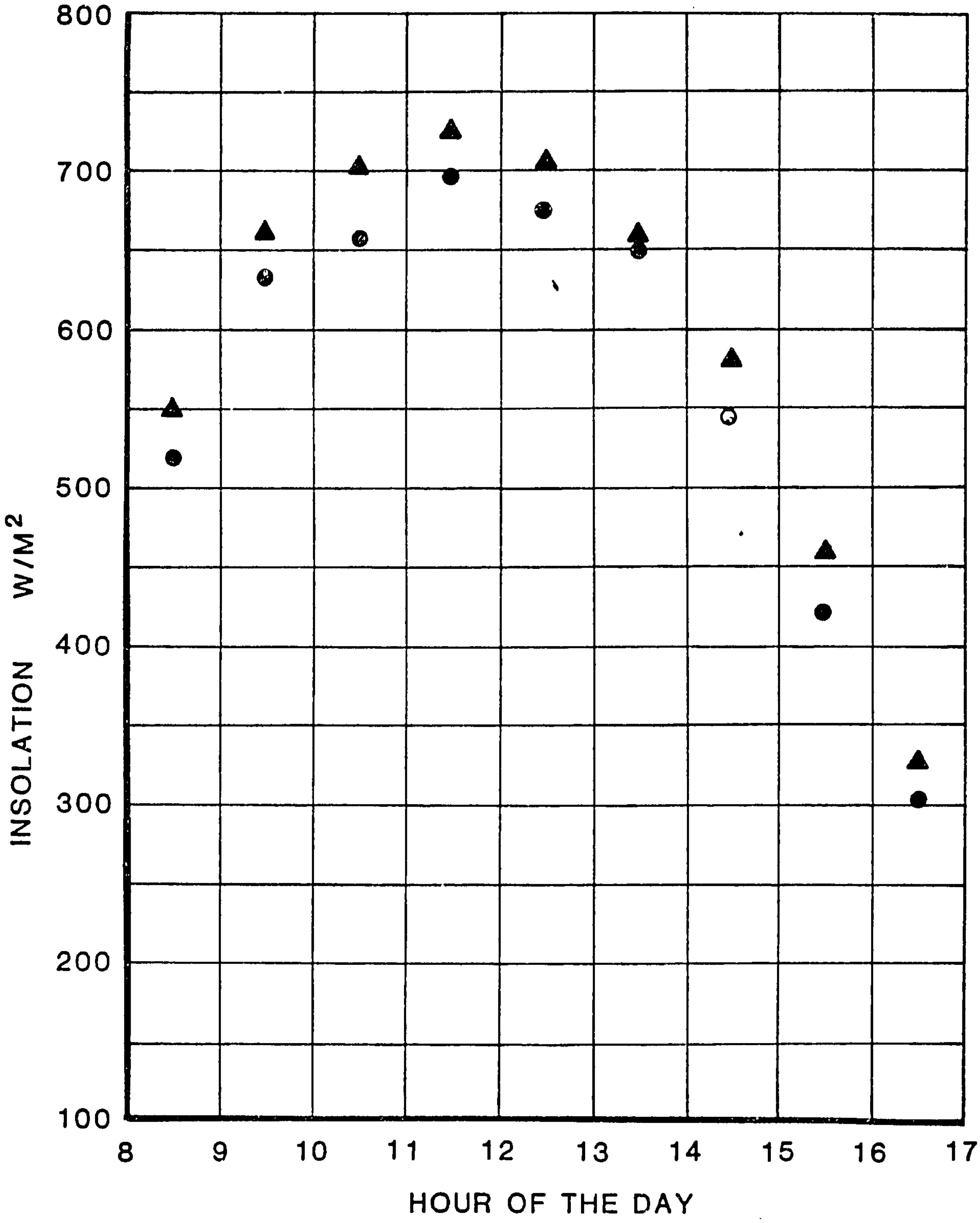


● – FOR ALL DAYS

▲ – FOR DAYS WITH 0.5 KW/M^2 INSOLATION FOR MINIMUM 5 HRS

FIG. 5 DIRECT SOLAR INSOLATION NORMAL TO DISH
MONTHLY AVERAGE

MAY,80
HYDERABAD

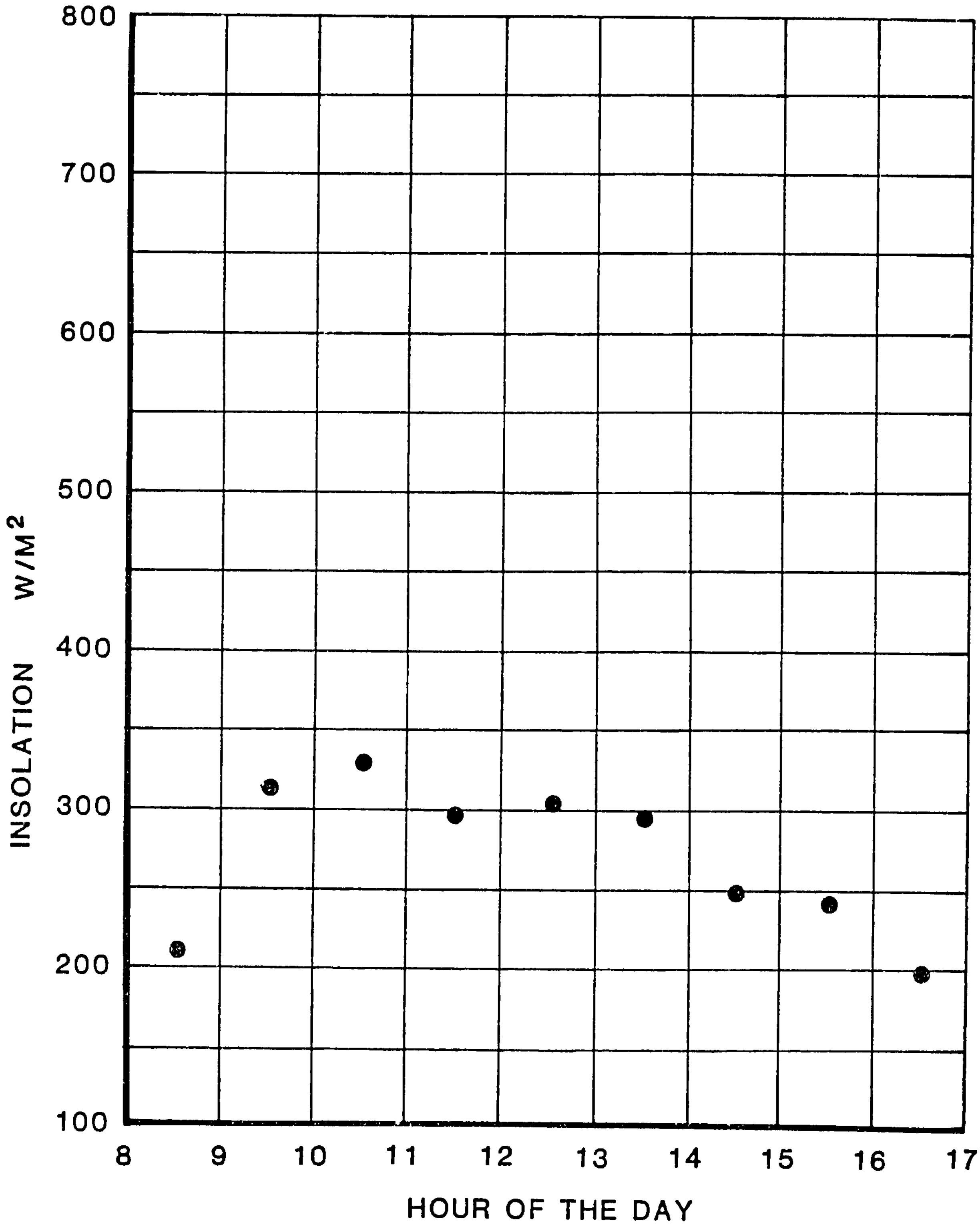


● – FOR ALL DAYS

▲ – FOR DAYS WITH 0.5 KW/M^2 INSOLATION FOR MINIMUM 5 HRS

FIG. 6 DIRECT SOLAR INSOLATION NORMAL TO DISH MONTHLY AVERAGE

JUNE,80
HYDERABAD



● – FOR ALL DAYS

FIG. 7 DIRECT SOLAR INSOLATION NORMAL TO DISH
MONTHLY AVERAGE

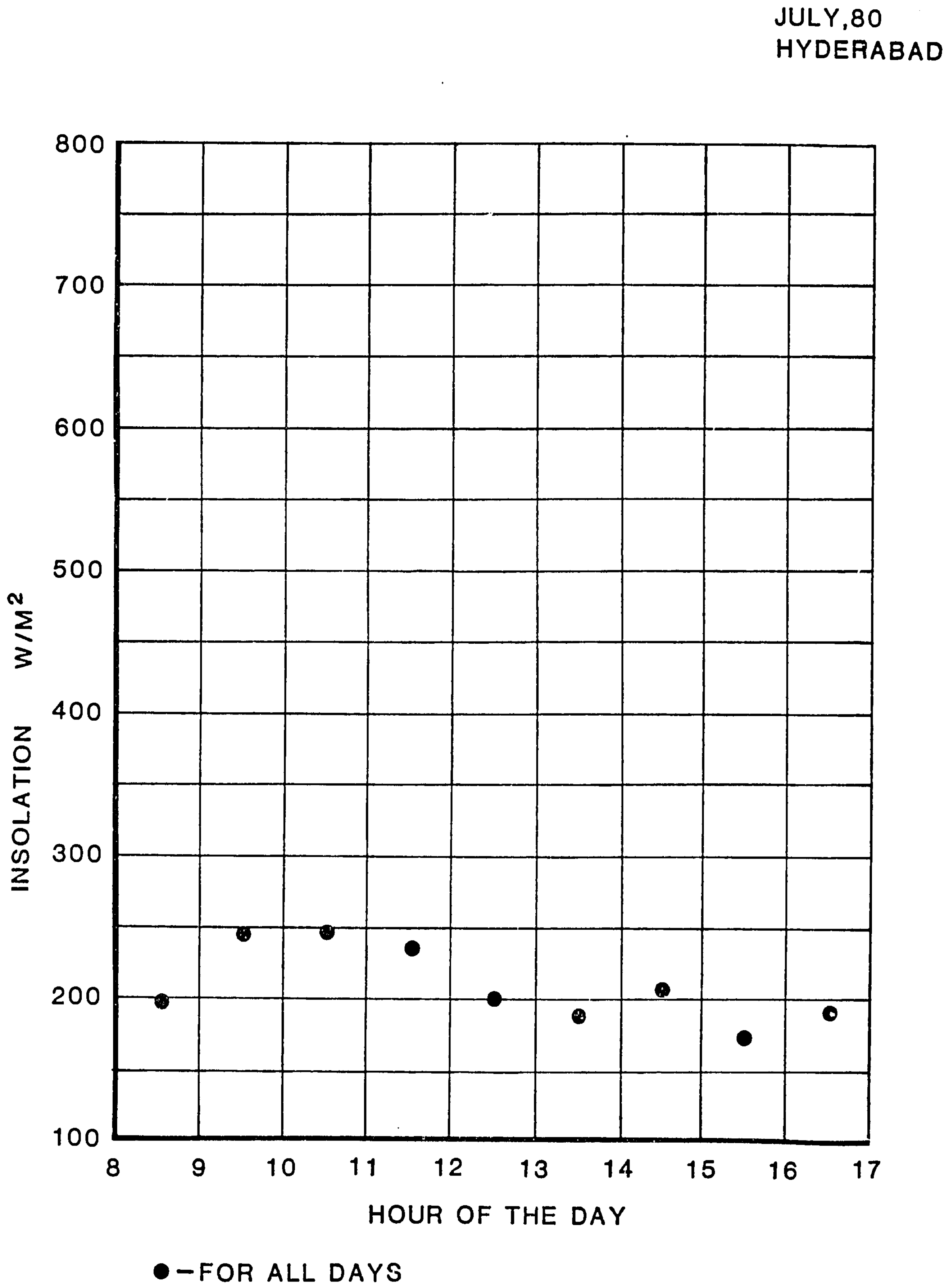
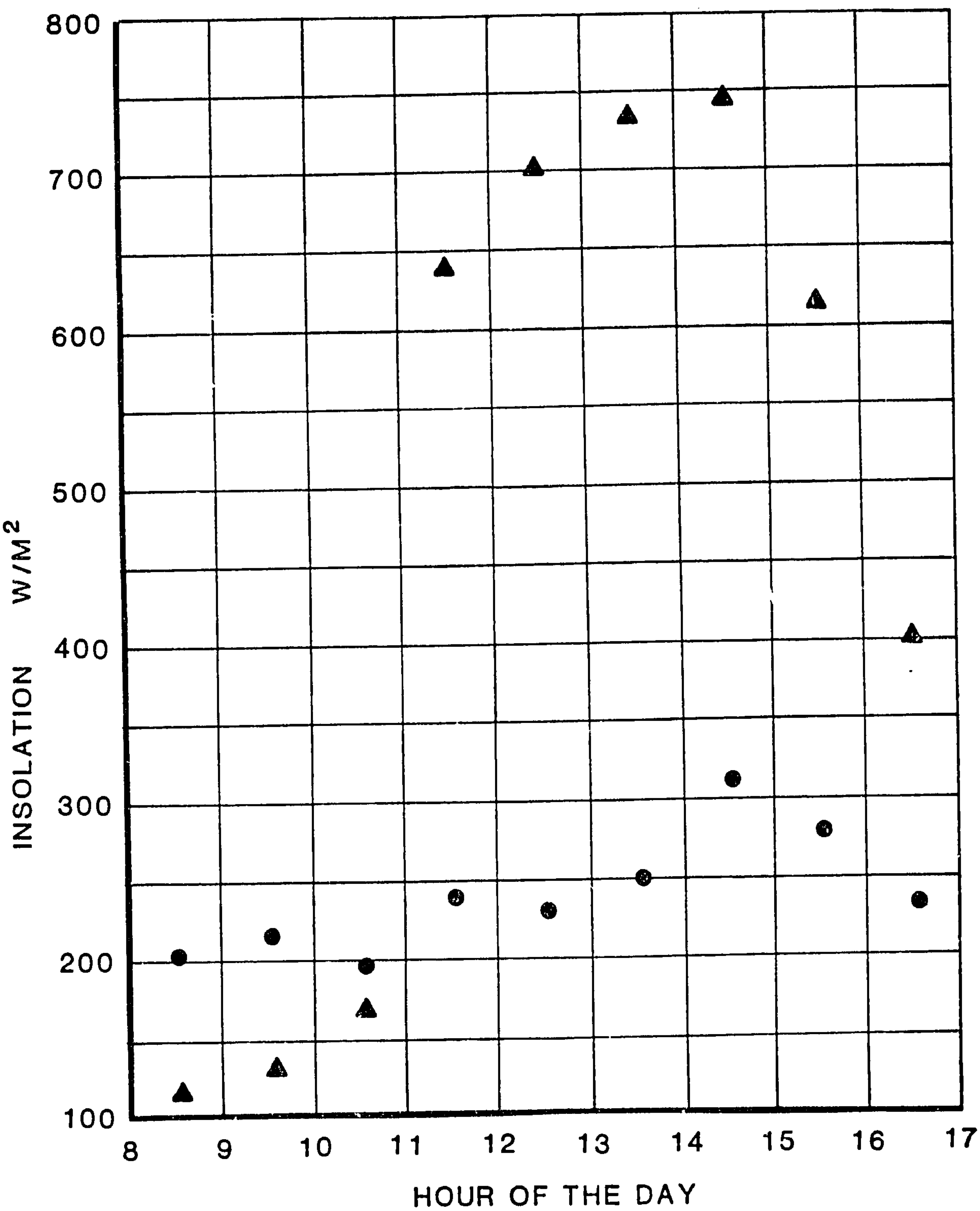


FIG. 8 DIRECT SOLAR INSOLATION NORMAL TO DISH MONTHLY AVERAGE

AUGUST,80
HYDERABAD



●—FOR ALL DAYS

▲—FOR DAYS WITH 0.5 KW/M^2 INSOLATION FOR MINIMUM 5 HRS

FIG. 9 DIRECT SOLAR INSOLATION NORMAL TO DISH MONTHLY AVERAGE

SEPTEMBER,80
HYDERABAD

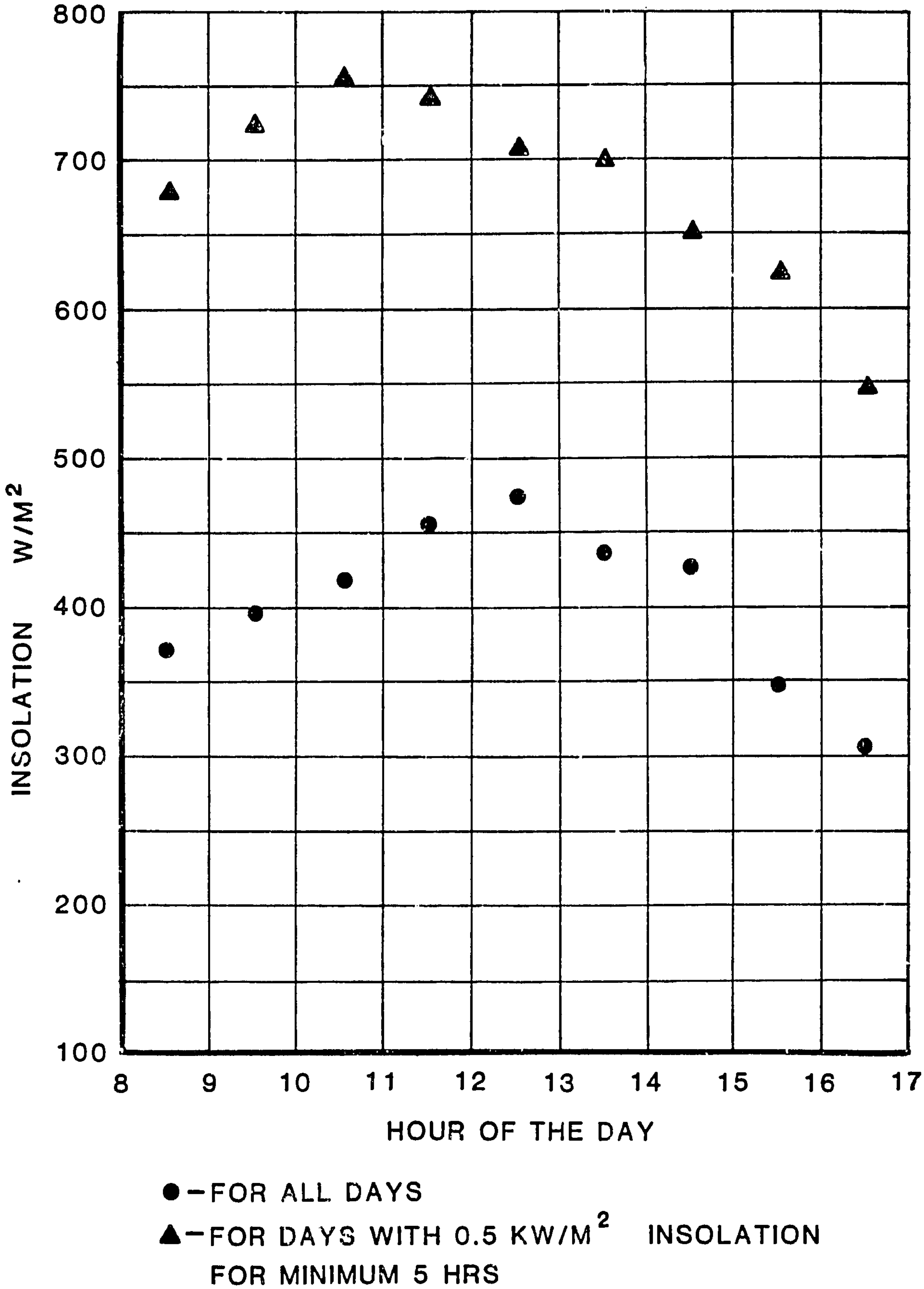
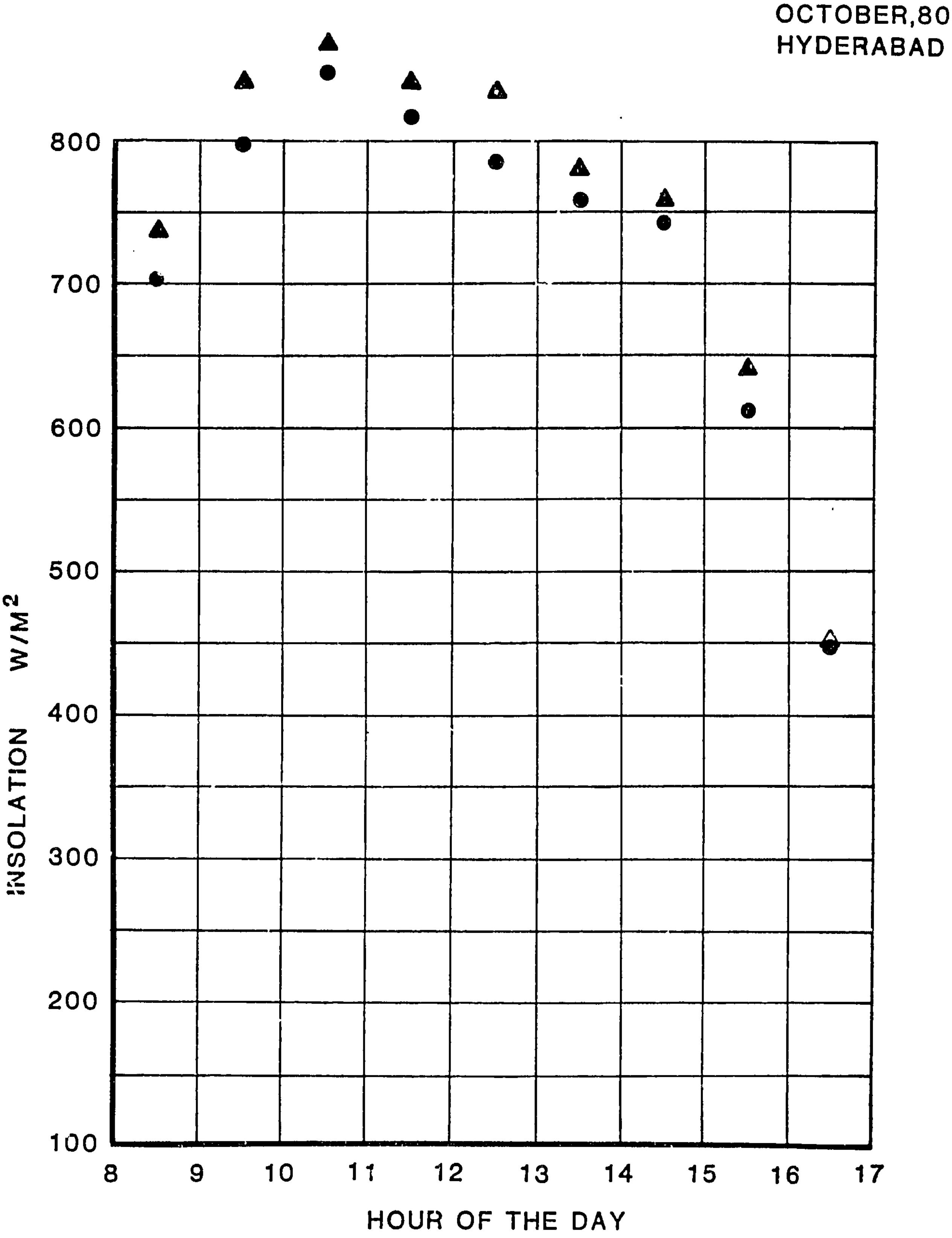


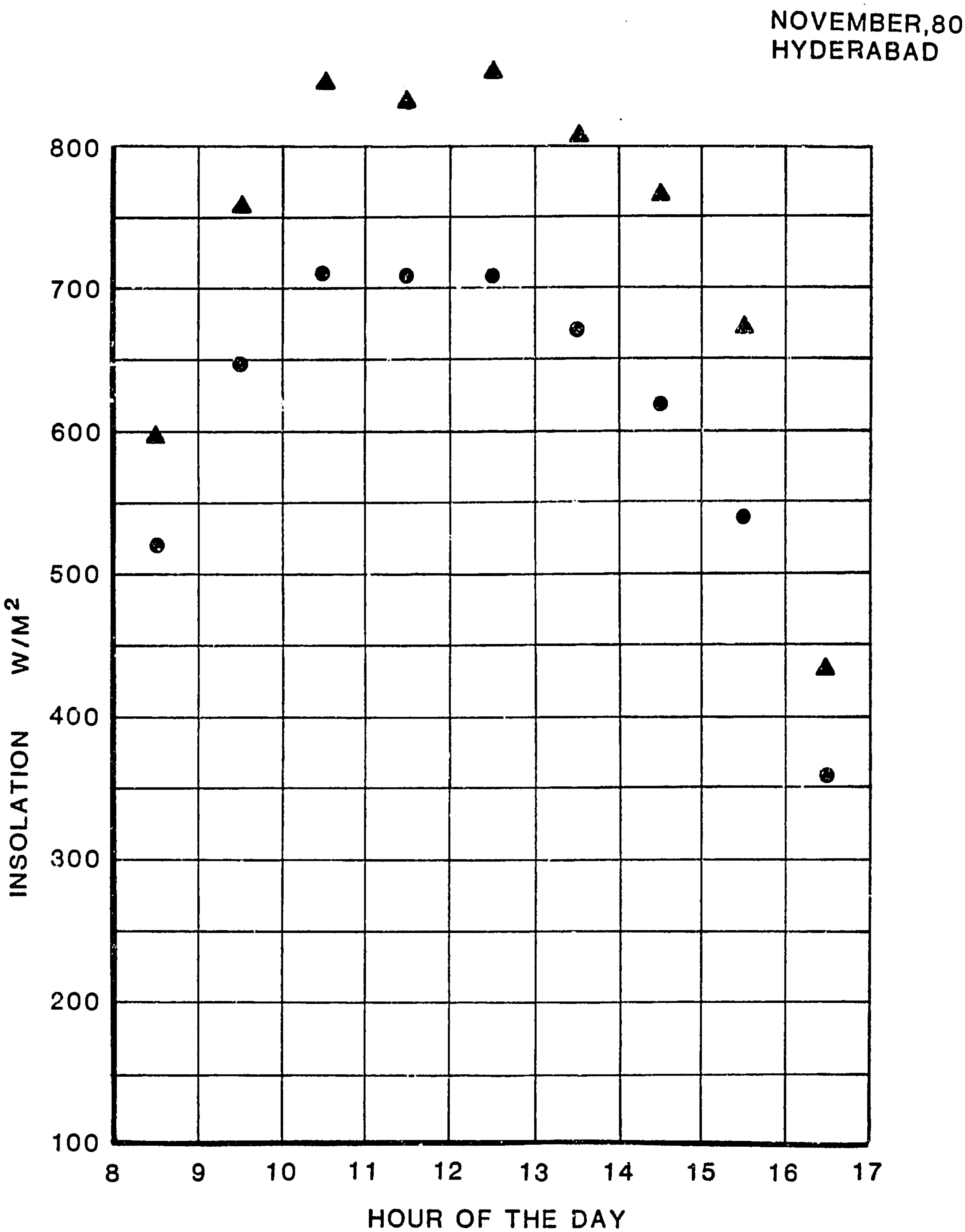
FIG. 10 DIRECT SOLAR INSOLATION NORMAL TO DISH
MONTHLY AVERAGE



● – FOR ALL DAYS

▲ – FOR DAYS WITH 0.5 KW/M^2 INSOLATION FOR MINIMUM 5 HRS

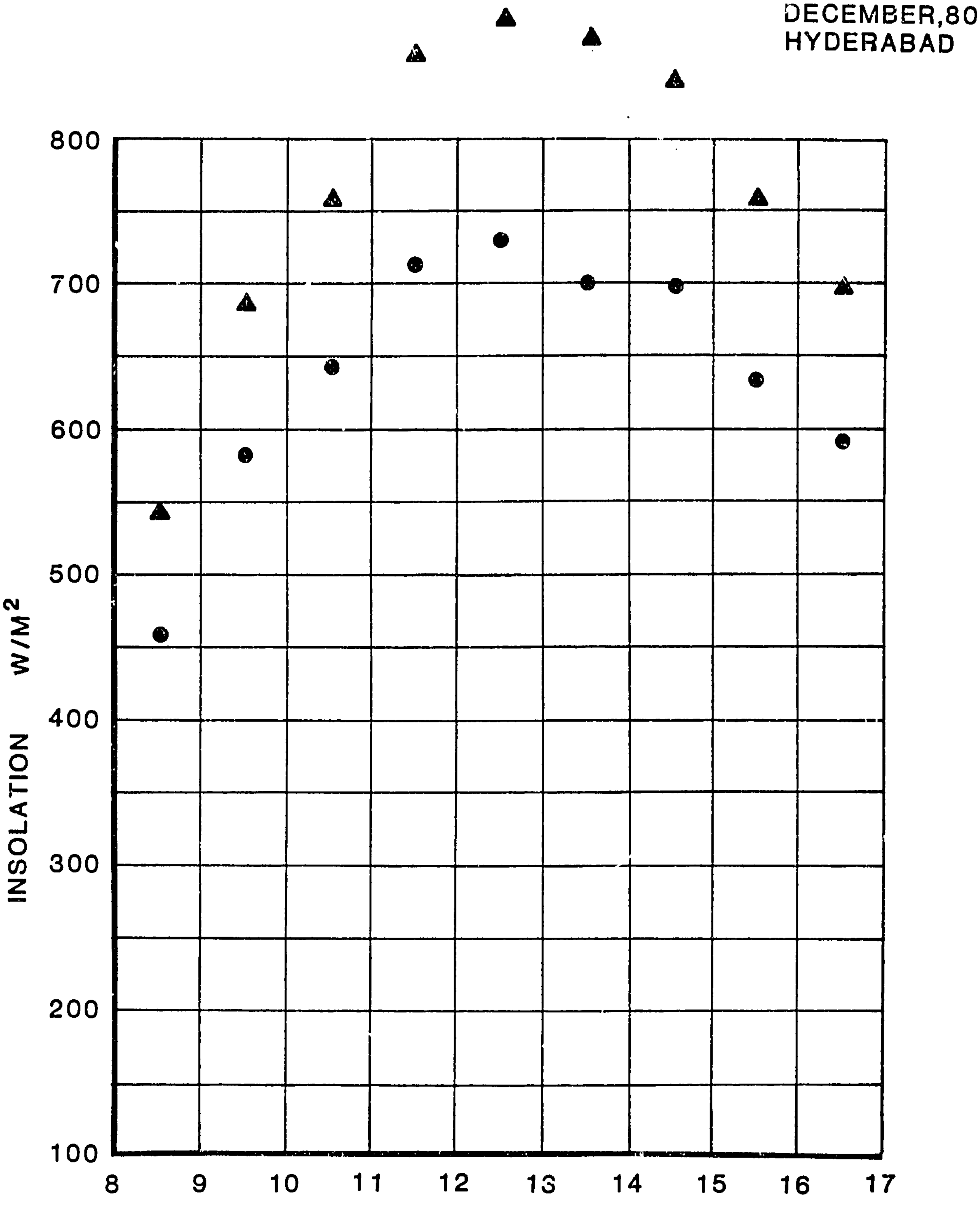
FIG. 11 DIRECT SOLAR INSOLATION NORMAL TO DISH
MONTHLY AVERAGE



● – FOR ALL DAYS

▲ – FOR DAYS WITH 0.5 KW/M^2 INSOLATION FOR MINIMUM 5 HRS

FIG. 12 DIRECT SOLAR INSOLATION NORMAL TO DISH
MONTHLY AVERAGE



●–FOR ALL DAYS
▲–FOR DAYS WITH 0.5 KW/M^2 INSOLATION FOR MINIMUM 5 HRS

MONTHS \ INSOLATION	0.50 kW/M^2	0.55 kW/M^2	0.60 kW/M^2
January	29	28	26
February	28	27	23
March	25	20	14
April	21	12	8
May	28	22	18
June	Nil	Nil	Nil
July	Nil	Nil	Nil
August	1	1	1
September	9	9	8
October	29	29	29
November	23	22	20
December	24	22	19
YEARLY	207	192	166

Table 4. NUMBER OF DAYS DIRECT SOLAR RADIATION NORMAL TO DISH With 0.50, 0.55, and 0.60 kW/M^2 for at Least 5 Hrs. a Day

MONTHS \ INSOLATION	0.75 kW/M^2	0.80 kW/M^2	0.85 kW/M^2	0.90 kW/M^2
January	27	16	7	1
February	12	1	Nil	Nil
March	3	Nil	Nil	Nil
April	3	Nil	Nil	Nil
May	13	10	5	2
June	Nil	Nil	Nil	Nil
July	Nil	Nil	Nil	Nil
August	Nil	Nil	Nil	Nil
September	9	6	4	Nil
October	30	29	27	15
November	20	18	15	12
December	24	24	22	15
YEARLY	141	104	80	45

Table 5. NUMBER OF DAYS DIRECT SOLAR RADIATION NORMAL TO DISH With 0.75, 0.80, 0.85, and 0.90 kW/M^2

of days with minimum 0.75, 0.8, 0.85 and 0.9 kW/m^2 direct solar isolation normal to dish. From these values, a peak insolation level of 0.8 kw/m^2 has been choosen. At this insulation level the plant is expected to deliver maximum power. However, the engine should be capable of taking care of those days with insolation more than 0.8 kW/m^2.

5.0 RANKINE CYCLE SYSTEM ANALYSIS

5.1 Conceptual System Description

The solar thermal power plant system uses dish collectors as the energy collection system and a steam engine as the prime mover, working on the Rankine cycle. The system can work on either the distributed or centrally located energy conversion mode. The collector will incorporate receivers capable of generating steam at the required conditions. The system does not include thermal storage for dark hours operation but has adequate buffer storage for short term transients. The system is expected to deliver its minimum output at 0.55 kW/m^2 insolation level, the maximum efficiency being at 0.80 kW/m^2 insolation. The auxiliary power required by the system is expected to be supplied through a biogas power generating system. Figure 13 represents the schematic diagram of the conceptual system for the centrally located power conversion unit. For the distributed power conversion system, the principle remains same, but all the components are located with the receiver and a air cooled condenser is preferred for this system. Because of the budget limitations, the output from the power plant has been limited to the order of 12 kW_e nominal.

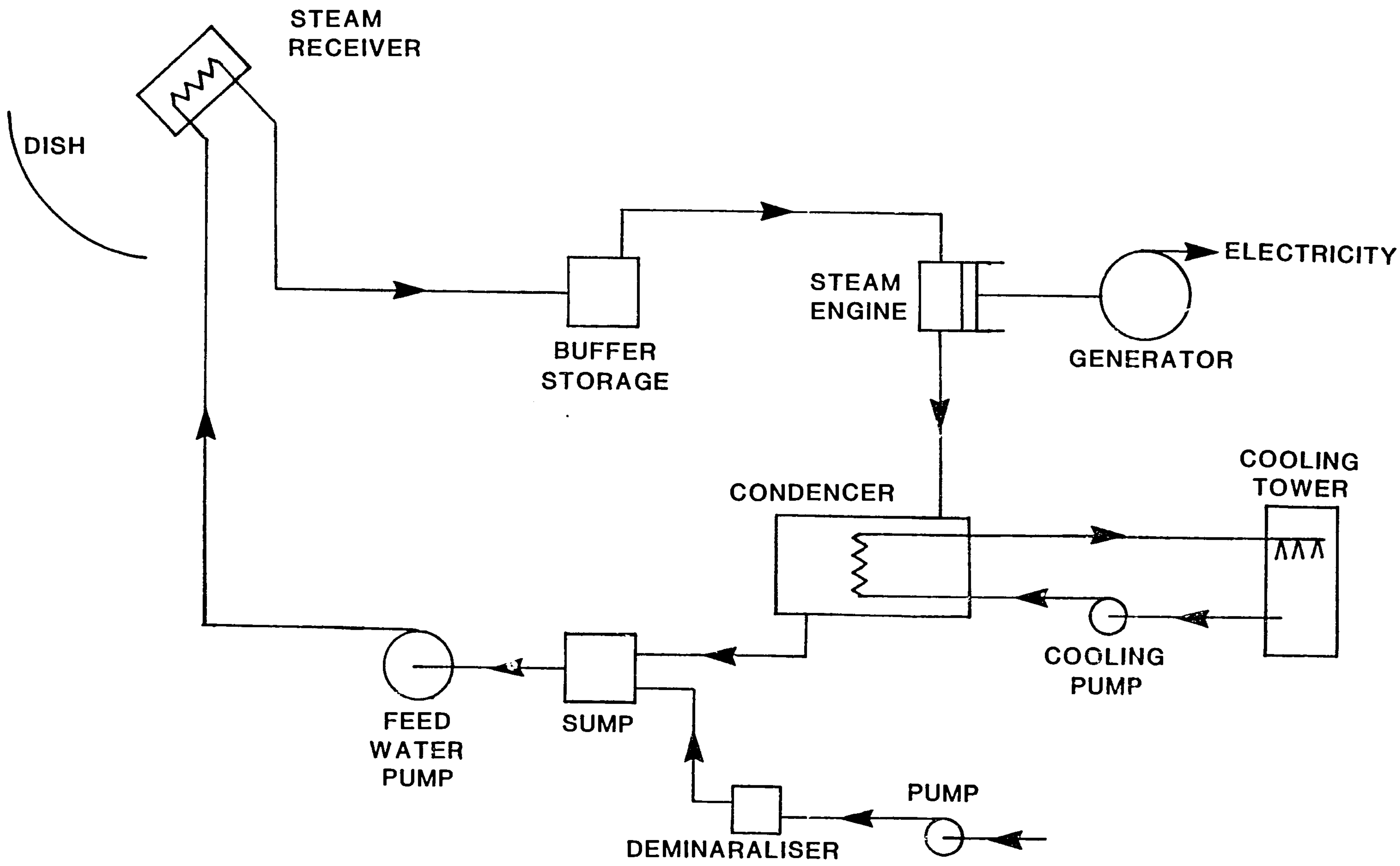


FIG. 13 SCHEMATIC DIAGRAM OF CONCEPTUAL SYSTEM

5.2 Finalization of Operating Parameters

One could analyze the steam Rankine cycle for solar operation and optimize the operating parameters for the best performance, then design and fabricate the most efficient subsystems/components matching the operating parameters. Thus, one could have the best possible solar power plant working on the steam Rankine cycle. In the present project, a steam engine is to be procured from the market and possible modifications are to be incorporated for the solar thermal operation. In this situation the evaluation of the operating parameters should be based on the characteristics of the available engine and not on the methodology explained above.

JPL has obtained two steam engines from Jay Carter Enterprises, Inc., Texas, which are under test for solar operations after carrying out necessary modifications. The capacities of the engines are, also, suitable for present system. The big Carter engine is suitable for the central conversion unit, where the Precursor Engine (small Carter Engine) is suitable for the distributed conversion unit. The specifications of both the engines, as envisaged by JPL for solar operation, have been provided in Table 6. Both being the possible candidates for the present system, the analysis has been carried out considering these two engines. If some other engine is adopted at a later date through further market survey, then the analysis has to be modified matching to the operating conditions of the new engine.

Table 6. SPECIFICATIONS OF CARTER ENGINES FOR SOLAR OPERATION

A) Precursor Engine:

Shaft Output	: 5 hp
Single Cylinder (uniflow)	
Bore/Stroke Ratio	: 2"/2.5" (50.8mm/63.5mm)
Expansion Ratio	: 10/1 (approx.)
Steam Temperature	: 1050°F (565.6°C)
Steam Pressure	: 1000-1500 psia (68-102 ata)
RPM	: 1800 (nominal))
Thermal Efficiency	: 15-20% (approx.)
Electric Power	: 3-5 kW_e (approx.)

B) Big Carter Engine:

Shaft Output	: 20-25 hp (nominal)
Two Cylinder (uniflow) Type	
Bore/Stroke Ratio :	: 2.5"/3" (63.5mm/76.2mm)
Expansion Ratio	: 11/1
Steam Temperature	: 1050°F (565.6°C)
Steam Pressure	: 1000-1500 psia (68-102 ata)
RPM	: 3600 (nominal)
Thermal Efficiency	: 20% (approx.)
Electrical Power	: 15-20 kW_e (approx.)

These engines are capable of taking very high pressure in the order of 170 ata and temperature around 695°C. Computer simulations on both these engines are available[1]. The programme has been used to carry out simulations of the present operating conditions. Table 7 presents the simulation results for the big Carter engine. One can find from Table 7 (h) that for an inlet temperature of 400°C and thermal input 130 kW, the engine can deliver 19 kW_e at 14.6% efficiency with an inlet pressure of 849 psia (57.75 ata) requiring a mass flow of steam of 370 lb/hr (167.9 kg/hr). The generator efficiency is 92% and overall conversion efficiency is fairly constant (varying from 14.4 to 14.6%) for the insolation range of 0.55 kw/m^2 to 0.80 kw/m^2. The major drawback of this engine is that the exhaust pressure is approximately 6 ata with a temperature of 157°C. However, since this condition is very close to saturation, there is no necessity of a feed water heater.

TABLE 7.

JAY CARTER ENTERPRISES STEAM ENGINE SIMULATION

A)

ENGINE TYPE	SIMPLE	ENGINE SPEED	3600. RPM
THERMAL INPUT	1.47. KW	STEAM TEMPERATURE	752. DEG.F
IDEAL CYCLE EFF	0.209	CONDENSING TEMP.	212. DEG.F
NO. OF CYLINDERS	2	EXPANSION RATIO	14.4/1
CYLINDER DIAMETER	2.500 IN.	DISPLACEMENT	29.452 CU.IN.
FRICTION LOSS	1.90 HP	THERMAL LOSS	1.36 HP

THERM	POW	NSHP	ACP	P1	H1	P2	H2	QW	TFW	EG	EX	EHE	EPC
146.90	25.0	36.5	1.54	1967.	1305.	87.	1056.	408.	109.	92.0	92.1	18.5	17.1
130.59	22.3	32.6	1.26	1824.	1314.	80.	1061.	361.	114.	92.0	91.2	18.7	17.2
114.29	19.8	28.9	1.02	1685.	1322.	73.	1067.	318.	119.	92.0	90.2	18.8	17.3
97.98	17.0	24.6	0.78	1530.	1331.	65.	1074.	270.	126.	92.9	88.6	18.7	17.4

B)

ENGINE TYPE	SIMPLE	ENGINE SPEED	3600. RPM
THERMAL INPUT	147. KW	STEAM TEMPERATURE	752. DEG.F
IDEAL CYCLE EFF	0.198	CONDENSING TEMP.	212. DEG F
NO. OF CYLINDERS	2	EXPANSION RATIO	10.0/1
CYLINDER DIAMETER	2.500 IN	DISPLACEMENT	29.452 CU.IN
FRICTION LOSS	1.90 HP	THERMAL LOSS	1.36 HP

THERM	POW	NSHP	ACP	P1	H1	P2	H2	QW	TFW	EG	EX	EHE	EPC
146.90	23.8	34.7	1.17	1410.	1338.	89.	1104.	410.	152.	92.0	91.7	17.7	16.3
130.59	21.4	31.2	0.97	1305.	1344.	81.	1108.	367.	156.	92.0	90.8	17.7	16.3
114.29	18.7	27.3	0.77	1199.	1350.	74.	1112.	321.	160.	92.0	89.6	17.7	16.3

C)

ENGINE TYPE	SIMPLE	ENGINE SPEED	3600. RPM
THERMAL INPUT	100. KW	STEAM TEMPERATURE	752. DEG.F
IDEAL CYCLE EFF	0.196	CONDENSING TEMP.	212. DEG.F
NO. OF CYLINDERS	2	EXPANSION RATIO	8.0/1
CYLINDER DIAMETER	2.500 IN.	DISPLACEMENT	29.452 CU.IN.
FRICTION LOSS	1.90 HP	THERMAL LOSS	1.36 HP

THERM	POW	NSHP	ACP	P1	H1	P2	H2	QW	TFW	EG	EX	EHE	EPC
100.00	15.3	22.4	0.59	934.	1365.	72.	1138.	280.	184.	92.0	87.6	16.8	15.4
88.90	13.6	19.8	0.48	876.	1367.	67.	1142.	250.	186	92.0	86.2	16.6	15.3
77.80	11.7	17.1	0.38	812.	1371.	62.	1146.	219.	190.	92.0	84.3	16.4	15.1
66.70	9.9	14.3	0.29	742.	1374.	56.	1151.	188.	195.	92.9	81.7	16.0	14.8
55.60	7.9	11.5	0.22	690	1377.	51.	1157.	158.	200.	92.1	78.3	15.4	14.2
44.50	5.9	8.6	0.16	630.	1380.	46.	1166.	127.	208.	92.8	72.8	14.3	13.3
33.40	3.8	5.6	0.10	569.	1383.	42.	1182.	97.	223.	91.1	63.7	12.4	11.3
22.30	1.2	1.7	0.06	478.	1387.	34.	1245.	66.	279.	91.0	35.2	5.8	5.3

TABLE 7.

D)

ENGINE TYPE	SIMPLE	ENGINE SPEED	3600. RPM
THERMAL INPUT	100. KW	STEAM TEMPERATURE	752. DEG.F
IDEAL CYCLE EFF	0.184	CONDENSING TEMP.	212. DEG.F
NO. OF CYLINDERS	2	EXPANSION RATIO	6.0/1
CYLINDER DIAMETER	2.500 IN.	DISPLACEMENT	29.452 CU.IN.
FRICTION LOSS	1.90 HP	THERMAL LOSS	1.36 HP

THERM	POW	NSHP	ACP	P1	H1	P2	H2	QW	TFW	EG	EX	EHE	EPC
100.00	14.4	21.1	0.49	714.	1376.	73.	1167.	286.	209.	92.0	86.9	15.7	14.4
88.90	12.7	18.5	0.39	665.	1378.	68.	1169.	253.	211.	92.0	85.3	15.5	14.3
77.80	10.9	15.9	0.31	608.	1381.	62.	1173.	222.	214.	92.0	83.3	15.3	14.0
66.70	9.2	13.3	0.22	548.	1384.	55.	1177.	190.	218.	92.9	80.5	14.9	13.8
55.60	7.3	10.7	0.18	514.	1385.	51.	1182.	160.	223.	92.1	76.9	14.2	13.1
44.50	5.4	7.8	0.12	466.	1387.	46.	1191.	127.	231.	92.8	70.9	13.1	12.2
33.40	3.4	5.0	0.08	425.	1389.	42.	1207.	97.	246.	91.1	60.7	11.1	10.1

E)

ENGINE TYPE	SIMPLE	ENGINE SPEED	3600.RPM
THERMAL INPUT	100. KW	STEAM TEMPERATURE	752. DEG.F
IDEAL CYCLE EFF	0.190	CONDENSING TEMP.	212. DEG F
NO. OF CYLINDERS	2	EXPANSION RATIO	7.0/1
CYLINDER DIAMETER	2.500 IN	DISPLACEMENT	29.452 CU.IN
FRICTION LOSS	1.90 HP	THERMAL LOSS	1.36 HP

THERM	POW	NSHP	ACP	P1	H1	P2	H2	QW	TFW	EG	EX	EHE	EPC
100.00	14.8	21.7	0.53	815.	1371.	73.	1153.	282.	197.	92.0	87.2	16 2	14.9
88.90	13.1	19.2	0.43	764.	1373.	68.	1156.	252.	199.	92.0	85.7	16.1	14.8
77.80	11.3	16.5	0.35	717.	1375.	64.	1159.	220.	202.	92.0	83.7	15.8	14.5
66.70	9.5	13.8	0.27	658.	1378.	58.	1164.	189.	207.	92.9	81.1	15.4	14.3
55.60	7.6	11.1	0.20	594.	1381.	52.	1170.	159.	212.	92.1	77.6	14.8	13.6
44.50	5.7	8.2	0.13	537.	1384.	46.	1179.	127.	220.	92.8	71.9	13.7	12.8
33.40	3.6	5.3	0.09	502.	1386.	43.	1195.	97.	234.	91.1	62.2	11.7	10.7
22.30	1.1	1.6	0.05	430.	1389.	36.	1258.	68.	291.	90.4	33.3	5.2	4.7

F)

ENGINE TYPE	SIMPLE	ENGINE SPEED	3600. RPM
THERMAL INPUT	120. KW	STEAM TEMPERATURE	752. DEG.F
IDEAL CYCLE EFF	0.182	CONDENSING TEMP.	212. DEG.F
NO. OF CYLINDERS	2	EXPANSION RATIO	6.0/1
CYLINDER DIAMETER	2.500 IN.	DISPLACEMENT	29.452 CU.IN.
FRICTION LOSS	1.90 HP	THERMAL LOSS	1.36 HP

THERM	POW	NSHP	ACP	P1	H1	P2	H2	QW	TFW	EG	EX	EHE	EPC
120.00	17.4	25.5	0.62	789.	1372.	82.	1163.	341.	205.	92.0	88.9	15.9	14.6
106.68	15.5	22.6	0.50	743.	1374.	77.	1165.	304.	208.	92.0	87.6	15.8	14.5
93.36	13.4	19.6	0.41	685.	1377.	70.	1168.	267.	210.	92.0	86.0	15.6	14.3
80.04	11.5	16.6	0.31	618.	1380.	63.	1172.	229.	214.	92.8	83.8	15.3	14.2
66.72	9.1	13.2	0.22	546.	1384.	55.	1177.	189.	218.	92.2	80.4	14.9	13.7
53.40	7.1	10.2	0.16	504.	1386.	50.	1183.	154.	224.	92.8	76.0	14.1	13.1
40.08	4.5	6.6	0.10	452.	1388.	44.	1197.	115.	236.	91.2	67.4	12.4	11.3
26.76	2.2	3.2	0.05	391.	1391.	38.	1227.	80.	263.	91.6	50.2	8.8	8.1

TABLE 7.

F)

ENGINE TYPE	SIMPLE	ENGINE SPEED	3600. RPM
THERMAL INPUT	147. KW	STEAM TEMPERATURE	752. DEG.F
IDEAL CYCLE EFF	0.179	CONDENSING TEMP.	212. DEG.F
NO. OF CYLINDERS	2	EXPANSION RATIO	6.0/1
CYLINDER DIAMETER	2.500 IN.	DISPLACEMENT	29.452 CU.IN.
FRICTION LOSS	1.90 HP	THERMAL LOSS	1.36 HP

THERM	POW	NSHP	ACP	P1	H1	P2	H2	QW	TFW	EG	EX	EHE	EPC
146.90	21.7	31.7	0.87	931.	1365.	98.	1158.	421.	201.	92.0	90.8	16.0	14.8
130.59	19.2	28.0	0.69	852.	1369.	89.	1160.	374.	203.	92.0	89.8	15.9	14.6
114.29	16.7	24.4	0.55	775.	1373.	80.	1163.	327.	206.	92.0	88.4	15.9	14.6
97.98	14.3	20.7	0.42	707.	1376.	73.	1167.	280.	209.	92.9	86.6	15.7	14.6
81.68	11.6	16.9	0.30	628.	1380.	64.	1171.	233.	213.	92.1	84.0	15.4	14.2
65.37	9.0	13.0	0.21	545.	1384.	55.	1177.	187.	219	92.6	80.2	14.8	13.7
49.06	6.2	9.1	0.13	488.	1386.	48.	1187.	141.	227.	91.2	73.8	13.7	12.5
32.76	3.2	4.7	0.07	414.	1390.	40.	1210.	94.	248.	91.9	59.4	10.8	9.9

H)

ENGINE TYPE	SIMPLE	ENGINE SPEED	3600.RPM
THERMAL INPUT	130. KW	STEAM TEMPERATURE	752. DEG.F
IDEAL CYCLE EFF	0.181	CONDENSING TEMP.	212. DEG F
NO. OF CYLINDERS	2	EXPANSION RATIO	6.0/1
CYLINDER DIAMETER	2.500 IN	DISPLACEMENT	29.452 CU.IN
FRICTION LOSS	1.90 HP	THERMAL LOSS	1.36 HP

THERM	POW	NSHP	ACP	P1	H1	P2	H2	QW	TFW	EG	EX	EHE	EPC
130.00	19.0	27.7	0.70	849.	1369.	89.	1161.	370.	203.	92.0	89.7	15.9	14.6
115.57	16.9	24.7	0.57	778.	1372.	81.	1163.	330.	206.	92.0	88.6	15.9	14.6
101.14	14.7	21.4	0.46	724.	1375.	74.	1166.	289.	209.	92.0	87.0	15.7	14.4
86.71	12.5	18.1	0.35	658.	1378.	67.	1170.	248.	212.	92.9	85.0	15.5	14.4
72.28	10.0	14.6	0.25	569.	1383.	57.	1175.	205.	216.	92.1	82.0	15.1	13.9
57.85	7.7	11.2	0.18	519.	1385.	52.	1181.	166.	222.	92.6	77.8	14.4	13.3
43.42	5.1	7.5	0.10	465.	1387.	46.	1192.	124.	232.	91.2	70.0	13.0	11.0
28.99	2.5	3.7	0.06	384.	1391.	37.	1220.	85.	257.	91.1	54.0	9.6	8.7

I)

ENGINE TYPE	SIMPLE	ENGINE SPEED	3600. RPM
THERMAL INPUT	130. KW	STEAM TEMPERATURE	842. DEG.F
IDEAL CYCLE EFF	0.195	CONDENSING TEMP.	212. DEG.F
NO. OF CYLINDERS	2	EXPANSION RATIO	6.0/1
CYLINDER DIAMETER	2.500 IN.	DISPLACEMENT	29.452 CU.IN.
FRICTION LOSS	1.90 HP	THERMAL LOSS	1.52 HP

THERM	POW	NSHP	ACP	P1	H1	P2	H2	QW	TFW	EG	EX	EHE	EPC
130.00	20.5	29.8	0.73	877.	1419.	90.	1190.	362.	230.	92.0	89.9	17.2	15.8
115.57	18.3	26.6	0.59	802.	1422.	82.	1193	324.	233.	92.0	88.8	17.1	15.8
101.14	15.8	23.1	0.47	742.	1424.	75.	1196.	284.	236.	92.0	87.3	16.9	15.6
86.71	13.5	19.5	0.36	677.	1427.	68.	1200	244.	239.	92.8	85.3	16.7	15.5
72.78	10.8	15.7	0.26	598.	1430.	60	1206.	202.	244.	92.2	82.3	16.2	14.9
57.85	8.2	11.9	0.17	526.	1433.	52.	1213.	162.	251.	92.6	77.9	15.4	14.3
43.42	5.5	8.1	0.10	474.	1435.	47.	1225.	122.	262.	91.2	70.6	14.0	12.7
28.99	2.7	4.0	0.06	405.	1438.	40.	1255.	84.	289.	91.1	54.6	10.4	9.4

WHERE:

THERM = THERMAL INPUT-KW	POW = ELECTRIC OUTPUT-KW_e
NSHP = NET SHAFT POWER-HP	ACP = ACCESSORY POWER-HP
P1 = INLET PRESSURE-PSIA	H1 = INLET ENTHALPY-BTU/LB
P2 = EXHAUST PRESSURE-PSIA	H2 = EXHAUST ENTHALPY-BTU/LB
QW = MASS FLOW RATE-LB/HR	TFW = FEEDWATER TEMP. - DEG.F
EG = GENERATOR EFFICIENCY-%	EX = EXPANDER EFFICIENCY-%
EHE = HEAT ENGINE EFFICIENCY -%	EPC = POWER CONVERSION EFFICIENCY-%

Allowing for the differences in the actual and simulated results for big Carter engine[2], a conversion efficiency of 13% is considered for the peak operating condition. The efficiency is expected to fall to 12% at the minimum load operation corresponding to the insolation level of 0.55 kW/m^2. On the basis of the similar considerations, the conversion efficiencies for small Carter engine are expected to be 11% and 9% respectively. The generator efficiency, in this case, is around 0.85. The performance of the small engine is considered at the receiver output conditions.

On the basis of the performance of the JPL test bed collectors and steam receivers, the following assumptions are made on efficiencies for the present operating conditions.

Efficiency of dish = 80%
Efficiency of receiver = 85% and
Thermal loss in the transport piping system[3] = 8%

i) Central Power Conversion System:

For the purposes of analysis, 19 kW_e output (refer to Table 7,h) has been considered, keeping a sufficient margin for operation above 0.80 kW/m^2 insolation.

Therefore, thermal energy input to the engine

$= 19/0.13 = 146.15\ kW_t$

Thermal energy output from the receiver

$= 146.15/0.92 = 158.86\ kW_t$

Thermal energy output from the dish or input to the receiver

$= 158.86/0.85 = 186.89\ kW_t$

Thermal energy input to dish

$= 186.89/0.80 = 233.61\ kW_t$

Area of collector required $= 233.61/0.80$

$= 292.02m^2$

Therefore diameter of dish =

19.3m for single dish
13.6m for two dishes
11.1m for three dishes
9.6m for four dishes
8.6m for five dishes
7.9m for six dishes

Thus, six dishes of 8m diameter and $50m^2$ area each have been adopted.

Therefore the output becomes

$= 300 \times 0.8 \times 0.8 \times 0.85 \times 0.92 \times 0.13$
$= 19.5\ kW_e$

The energy output at $0.55\ kW/m^2$ insolation

$= 300 \times 0.55 \times 0.8 \times 0.85 \times 0.92 \times 0.12$
$= 12.4\ kW_e$

ii) Distributed Power Conversion System:

Power output from each dish

$= 50 \times 0.80 \times 0.80 \times 0.85 \times 0.11$
$= 3.0\ kW_e$

Therefore, total power output

$= 18.0\ kW_e$

Output at $0.55\ kw/m^2$ insolation

$= 50 \times 0.55 \times 0.80 \times 0.85 \times 0.09 \times 6$
$= 10.0\ kW_e$

Therefore, for a power plant size of 20 kW_e, the central power conversion system seems to be efficient in comparison with distributed power conversion system so far as Carter engines are concerned. The distributed system has some advantages over the central system in that the thermal mass is less due to a much smaller thermal transport system. This reduced mass permits a faster warm-up time and a higher kWh output. Also, the close coupled system prevents the thermal loss exhibited by the central system. The disadvantages of the distributed system are, an increased and complicated maintenance programme, and the system will require development and a thorough testing programme. The problems of weather proofing, vibration, oil circulation, and cooling have not been addressed. These problems, and the fact that there are six engines, would result in a considerable increase in cost and time. Therefore, for the first solar thermal power plant system, the central approach has been adopted and the following analysis has been carried out for this system only. The operating parameters for the system have been confined to comparatively low operating levels, at the cost of performance, for minimizing the problems of high temperature and pressure operations. However, this being the first prototype, care may be taken while designing and fabrication of components/subsystems to retain sufficient flexibility to carry out high temperature and pressure tests in order to gain experience at a later stage.

5.2.1 Operating Parameters for Big Carter Engine

Insolation	:	0.80 kW/m^2
Inlet pressure	:	60 ata (882 psia) (approx.)
Inlet temperature	:	400°C (752°F)
Outlet pressure	:	6 ata (88.2 psia) (approx.)
Outlet temperature	:	157°C (314.6°F)
Mass flow of steam	:	180 kg/hr (396.7 lb/hr)
Energy input	:	150 kW_t
Expansion ratio	:	6/1 (approx.)
RPM	:	3600
Overall output	:	19.5 kW_e
Generator efficiency	:	0.9 (approx.)
Conversion efficiency	:	0.13 (approx.)
Shaft output	:	21.7 kW_m with a maximum of 25 kW_m

5.2.2 Parameters for Dish Design

Diameter of dish reflecting surface: 8 m

Area of each dish reflecting surface: $50m^2$

Number of dishes: 6

Reflecting surface: Faceted mirrors of spherical surface with suitable radius of curvature

Dish efficiency: 0.8 (approx)

Focal length/dish diameter: 0.6

Two options are to be considered during design as follows.

Depending upon the cost and suitability to Indian fabrication facilities, one version has to be opted.

A) Polar mounted dish with parabolic or flat base and either with square or trapizoidal facet sections.

B) Flat circular helio dish with either square or trapizoidal facet sections.

Figures 14, 15A and 15B show the pictorial view of the above options.

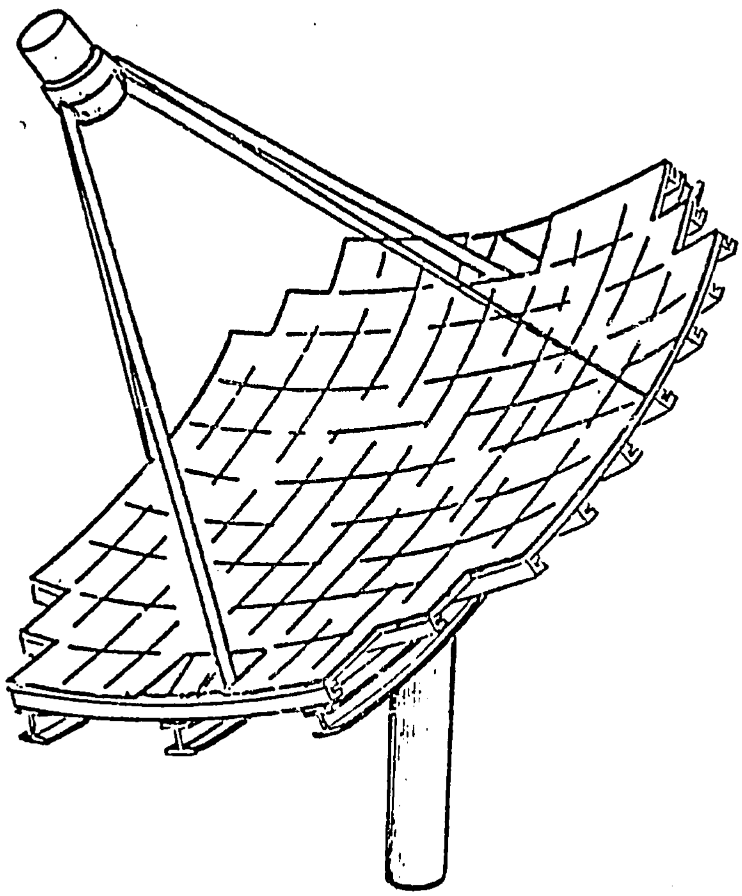


Figure 14. POLAR MOUNTED PARABOLIC DISH WITH SQUARE FACETS

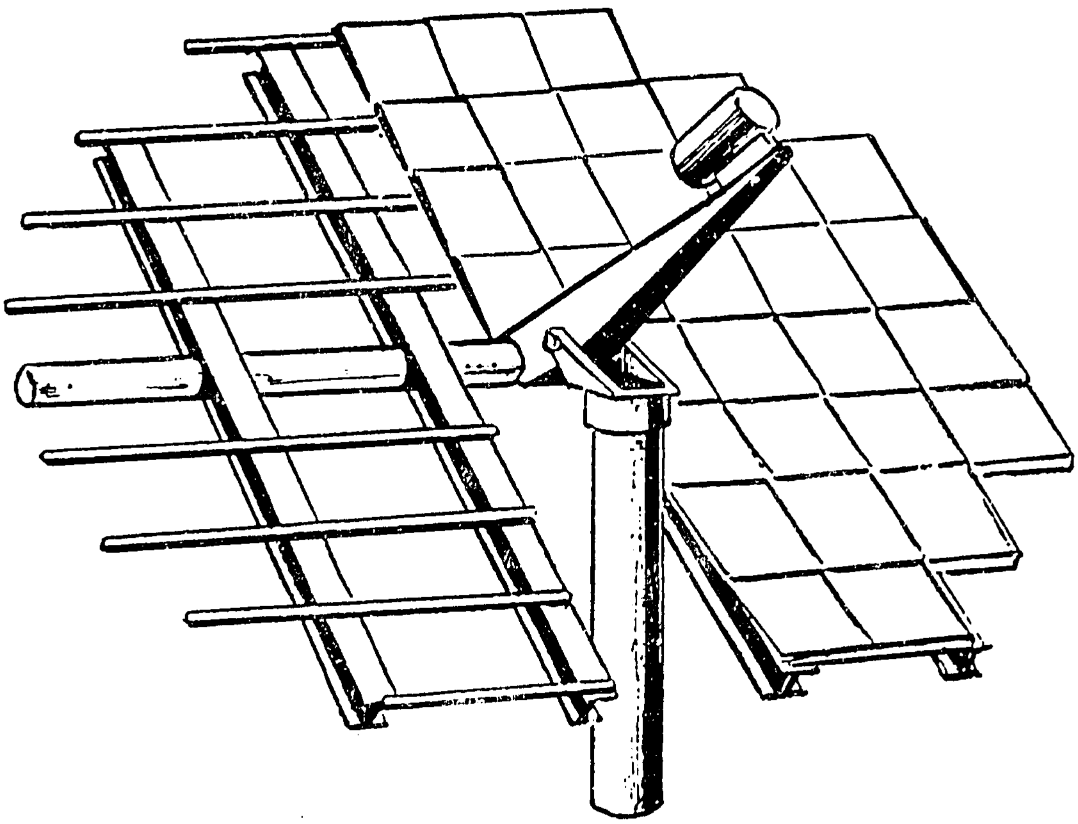


Figure 15 (A). FLAT HELIODISH WITH SQUARE FACETS

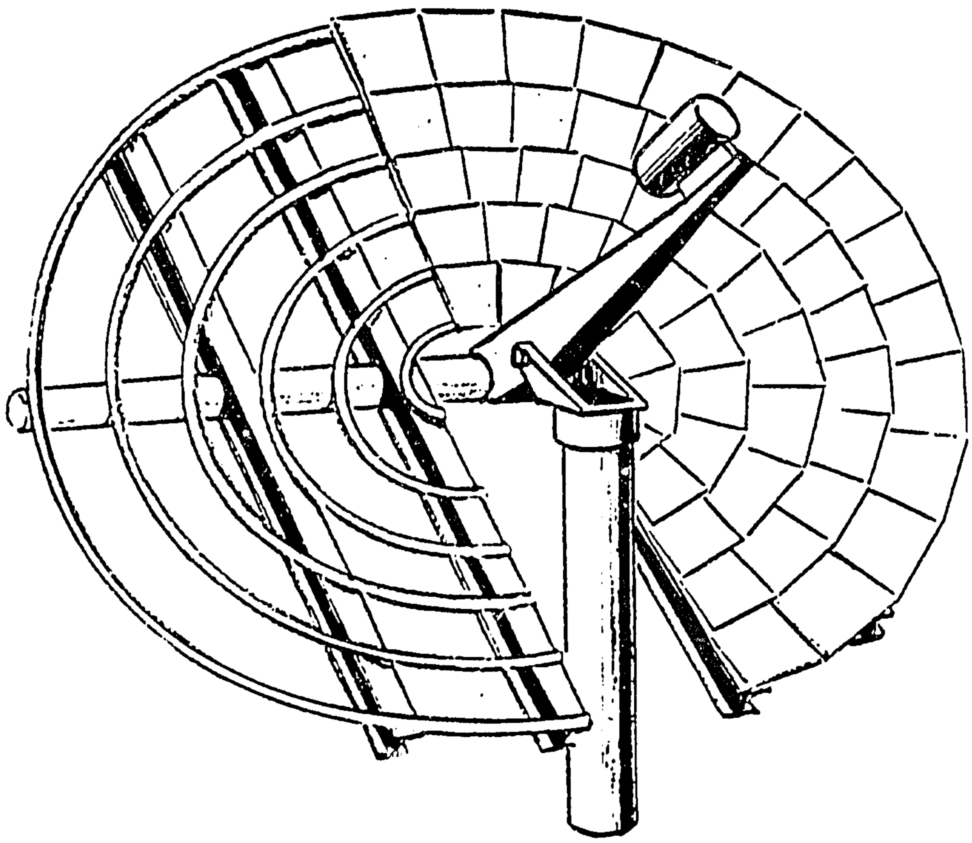


Figure 15 (B). FLAT HELIODISH WITH TRAPEZOIDAL FACETS

It is expected that the base area for flat contruction has to be more in comparison to dish to accomodate facets of higher curvetures, minimizing blockage and shading effects. The facets are of foam glass base with top reflecting mirrors of approximately 2 mm thickness. From the view point of material wastage, facets with square sections are preferable. It is desirable to have a balanced structural system in order to minimize the size of the tracking mechanism as well as input power to the tracking electronics. This can be achieved efficiently by pivoting the whole structure at the centre of gravity of the concentrator and receiver. Figure 16 shows a balanced system based on this principle. Balancing can, also, be obtained by providing cantilever weight on the back side of the receiver. In this case, the structure will be heavier. Various errors namely optical error (slope error), structural deflection error, pointing error and tracking errors etc., affecting the size of image should be evaluated and kept to a minimum possible value (in the order of 1.5 to 2.5 milliradian) to have a suitable receiver aperture.

5.2.3 Parameters for Receiver Design

Temperature:	Inlet 95°C (203°F) Outlet 500°C (932°F)
Pressure :	Inlet 70 ata (1029 psia) Outlet with minimum possible pressure drop
Type :	Cavity type with coil heat exchanger
Tube size :	3/8" (9.5mm) or 1/2" (12.7mm)OD stainless tube is preferred
Steam flow :	30 kg/hr (66.1 lb/hr)
Insolation :	0.8 kW/m^2
Power Input:	32 kW_t
Receiver efficiency:	0.85 (approx.)
Receiver aperture:	8" (200 mm) (approx.)

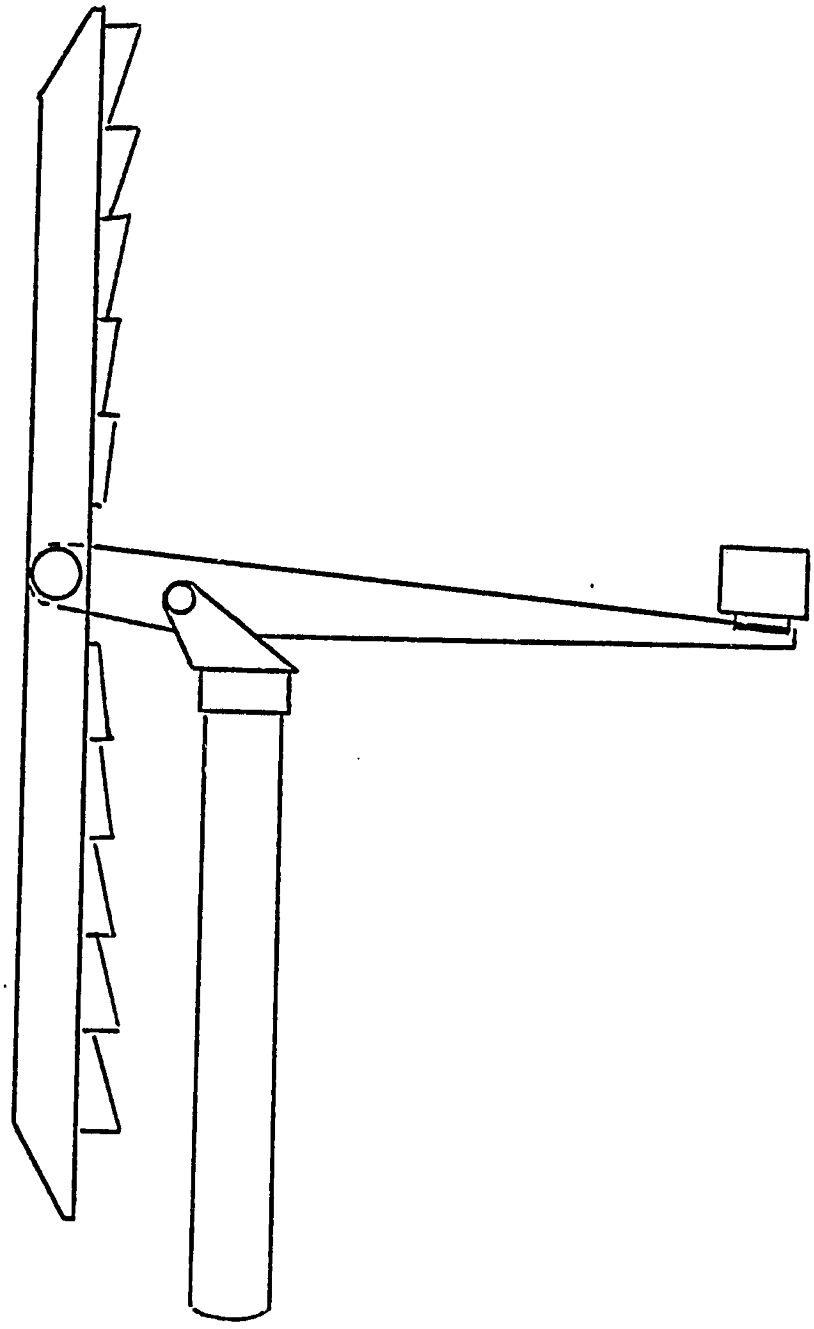


Figure 16. VIEW OF A BALANCED STRUCTURAL SYSTEM

The outlet temperature of the receiver has been decided after taking into account an 8% thermal loss in the steam transport system. To attain the desired engine inlet conditions, the inlet temperature to the receiver should have been about 150°C, but 95°C has been considered to be a design value which can accomodate an engine capable of expanding up to atmospheric pressure without further modifications in the heat exchanger coil of the receiver.

The receiver should be capable of delivering steam at 500°C (932°F) at 0.55 kW/m^2 insolation, with appropriate variation in mass flow. With a suitable aperature of the receiver matched to the dish size and faceted mirrors, two design options are to be considered as follows:

A) The heat exchanger tube may be of helical configuration with proper coil to tube diameter ratio, and

B) The heat exchanger tube may be of spiral configuration placed on the rear cover of the receiver.

Out of the above two, the best one from cost and performance considerations has to be chosen for final design and fabrication, with due consideration of flow stability. The design of the receiver should specify the following, if possible:

i) The reduction of flow in the heat transfer coil when insolation level falls, analyzed in step sizes of 0.05 kW/m^2, from 0.8 kW/m^2 to 0.3 kW/m^2 to attain constant outlet temperature of 500°C (932°F)

ii) Reduction in flow to attain higher temperature in steps of 25°C (47°F) from 500°C (932°F) to 550°C (1022°F) with 0.8 kW/m^2 insolation.

iii) Increase in flow to maintain the temperature of 500°C (932°F) for rise in insolation level from 0.8 kW/m^2 to 1.0 kW/m^2, analyzed in step sizes of 0.05 kW/m^2.

5.3 Steam Engine and Generator Assembly

The schematic flow diagram for the steam engine has been shown in Figure 16. The condensate pump (a gear pump) sucks condensate and oil mixture from condenser and supplies to oil separater where oil is separated and subsequently fed to the engine with the help of a oil pump, but water is drained into the sump. A feed water pump (reciprocating type) pumps water from the sump to six receivers which are in parallel.

The engine couples to a suitable generator for delivering three-phase electric output with constant voltage and frequency. This condition is very simple to attain when engine operates with constant speed. In the present case, the engine is expected to start from part load operation, attain peak level, and then go down to part load output corresponding to varying insolation. It may not be possible to get a constant rpm from an engine under these operating conditions. Hence, suitable arrangements must be incorporated to generate a constant three-phase electric output. Three options, in this regard, have been presented in Figure 17 for consideration and one will be selected on the basis of cost and performance.

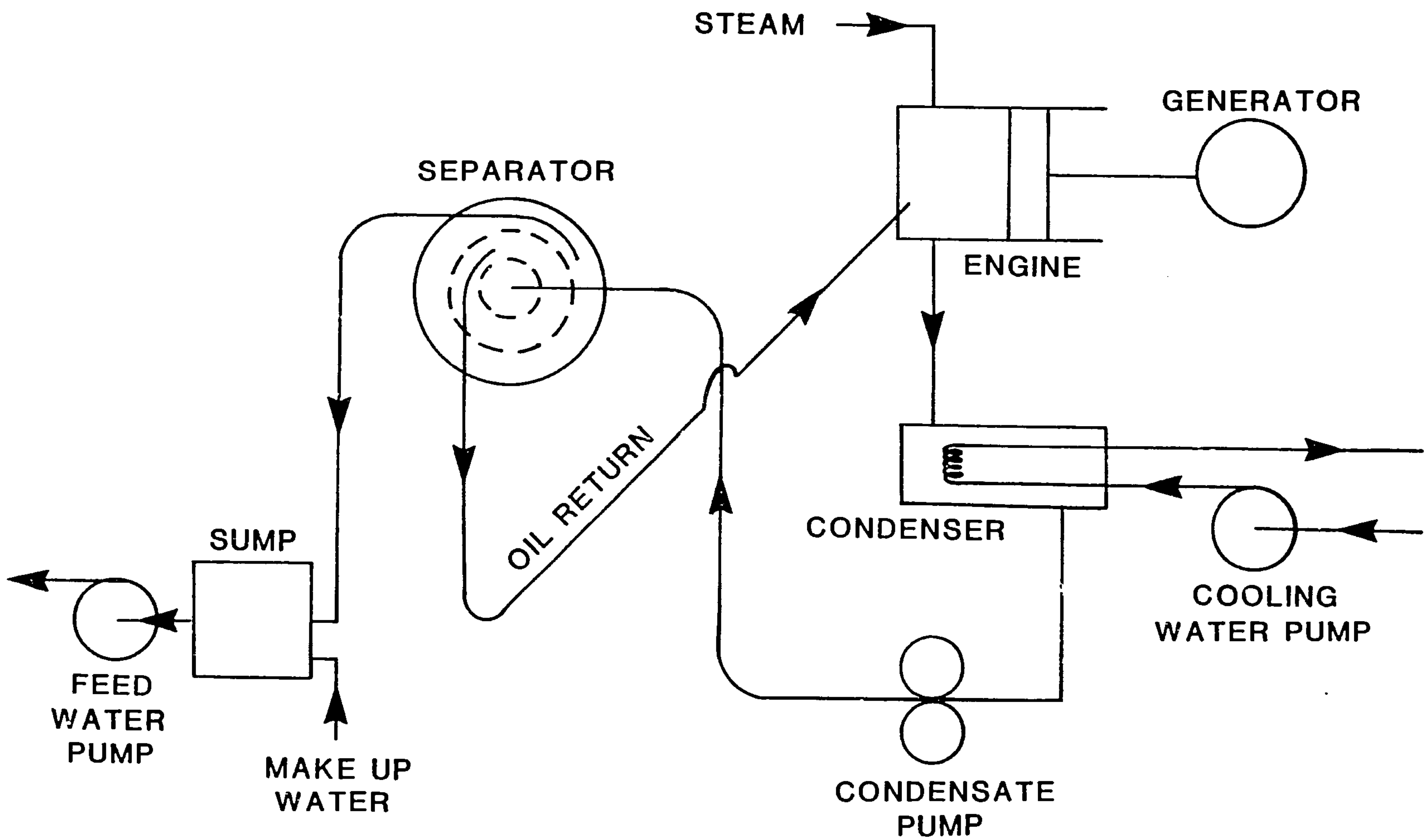


FIG.17 STEAM ENGINE FLOW DIAGRAM

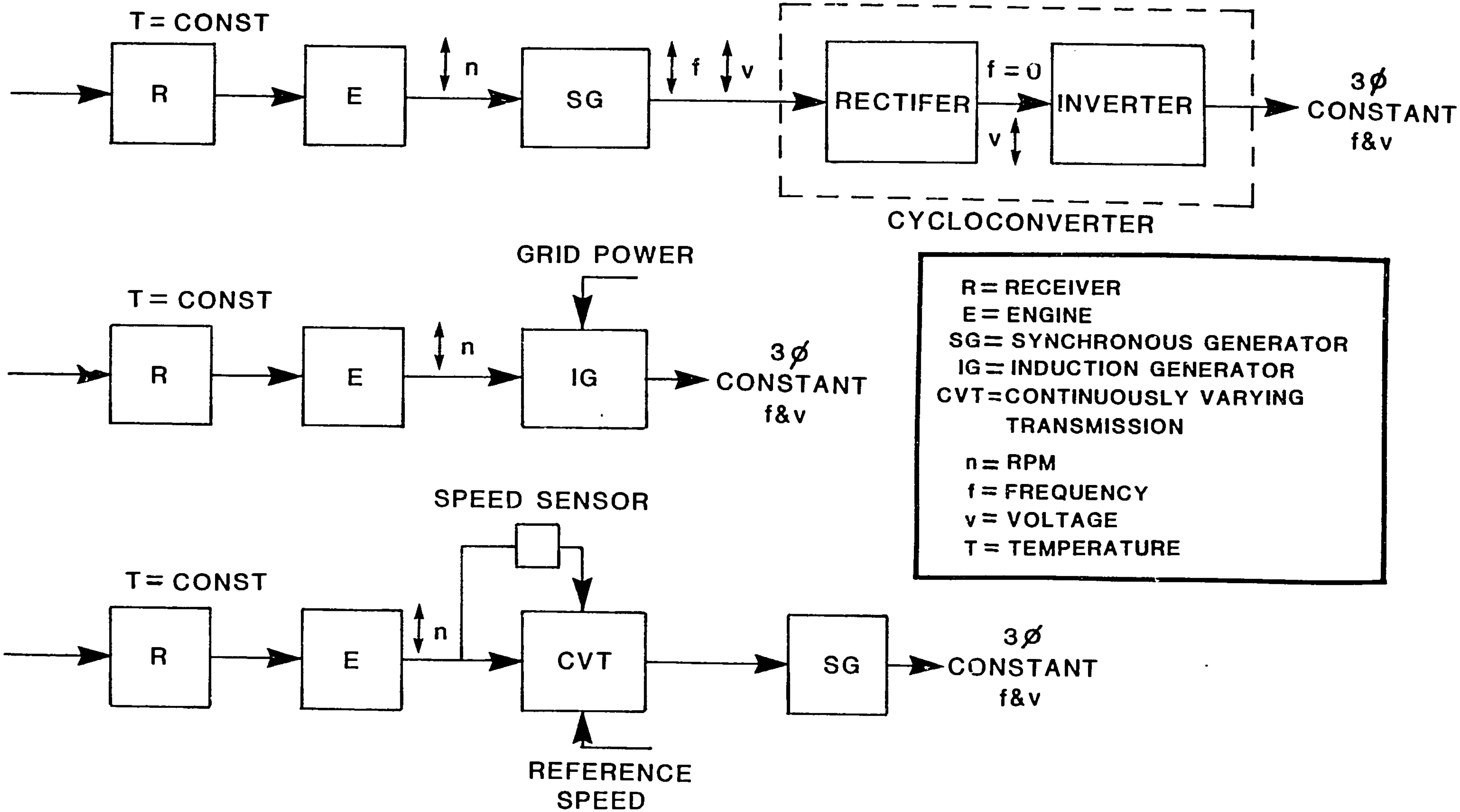


FIG.18 DESIGN OPTIONS FOR THE GENERATION OF A CONSTANT THREE PHASE OUTPUT WITH VARYING ENGINE SPEED

5.4 Thermal Design of the Condenser

Since the exhaust temperature from the steam engine is about 157°C, the system can use either a air cooled or water cooled condenser. This steam engine was developed basically for automobiles. Therefore, air cooled consensers are available in the market suitable for this engine. During testing of the engine at JPL, an air cooled condenser will be used. Therefore, design of the air cooled condenser has not been considered. The design of a water cooled condenser has been made and the design details have been presented in Appendix A. The salient features of the condenser are as under:

Hot fluid	:	Steam
Pressure	:	Inlet 6 ata Outlet 6 ata (approx.)
Temperature	:	Inlet 157°C Outlet 150°C
Mass flow	:	180 kg/hr.
Cold fluid	:	water
Temperature	:	Inlet 33°C Outlet 37°C
Mass flow	:	22185 kg/hr.
Heat transfer load		
Condensation	:	87120 kcal/hr.
Subcooling	:	1620 kcal/hr.
Condenser details		
Type	:	Horizontal, shell and fixed tube
Heat transfer area	:	9.4 ft^2 (0.87m^2)
Shell	:	6" (152.4mm) ID
Heat exchanger tube	:	3/4" (19mm)OD, 1.75'(0.533m) long, 14 BWG gauge
Number of passes	:	Single
Nozzle Size	:	Steam, 2" ID pipe Condensate, 3/8" ID pipe Cooling water, 3" ID pipe
Pressure drop	:	Tube side, 1.5 psi (0.1 ata) Shell side, negligible

5.5 Buffer Storage

Since insolation varies with diurnal and seasonal cycles as well as with local cloud cover, the thermal energy entering the receiver is subject to corresponding time-dependent fluctuations. The heat removed from the receiver also fluctuates. It has been indicated earlier that the present system will not incorporate any thermal storage to take care of diurnal and seasonal variation of insolation. The rapid or transient fluctuations due to cloud cover can be attenuated by using buffer storage. Depending on the capacity, the buffering reduces engine part load operation, improves efficiency, and alleviates control requirements. Buffer storage also reduces engine start/stop cycles, resulting from cloud cover, which is an important factor in extending engine life. Each power conversion unit requires a different amount of thermal buffering to handle variations in short-term insolation. The buffering can be provided either by latent heat storage or by storage of steam.

Latent Heat Buffer Storage:

The latent heat storage can be, conveniently, provided in the receiver for buffering between the variations in solar flux and the heat delivered from the receiver. The degree of buffering provided by the receiver depends upon its thermal inertia, heat flow paths, and control logic. Thermal buffering characteristics of the receiver containing phase change material (PCM) thermal energy storage, are evaluated under selected insolation patterns and different fluid flow control conditions to obtain the transient response of the receiver and corresponding fluid outlet temperatures. Therefore, transient response characteristics of buffering should take account of receiver characteristics, thermal capacity, flow control and internal heat transfer, etc.[4,5].

Table 8 shows the PCM candidates which have been selected for detailed investigations. In this report only simplified calculations on the requirement of the PCM material have been made.

Table 8. Selected PCM Candidates

	Salt Composition (by weight)	Melting Point (°C)	Heat of Fusion (Kcal/kg)	Containment Material
1	LiBr	547	49	SS 316 & 321
2	24 KCL-47BaCl$_2$ -29 CaCl$_2$	551	52	SS 316 & 321
3	20 Li$_2$CO$_3$-60 Na$_2$CO$_3$ -20 K$_2$CO$_3$	550	68	SS 316 & 321
4.	22 Li$_2$CO$_3$-16 Na$_2$CO$_3$ -62 K$_2$CO$_3$	550	69	SS 316 & 321

Capacity of buffer storage: 15 minutes (considered)

Temperature of receiver fluid: 500°C

Energy input to the receiver: 32 kW_t

Buffer storage requirement: $32 \times 15/60 = 8\ kW_t$

Weight of PCM salt (22 Li_2CO_3-16 Na_2CO_3-62 K_2CO_3) required: $8 \times 860/69 = 99.7$ kg

Density of this salt: 1625.6 kg/m^3

Volume of salt: 0.06133m^3
= 61331.2 cm^3

Table 9 gives the quantity of this PCM salt required for various storage duration. If this system is used, then one has to take care, while designing the base structure for receiver, the additional load due to the PCM material and its container. However, this type of buffer storage should be used with prior knowledge of thermal cycling characteristics including life cycling and corrosion behavior of the salt.

Table 9 Weight of PCM (22 Li_2CO_3-16 Na_2CO_3-62 K_2CO_3) Salt for Various Storage Duration

Weight of PCM salt (kg)	66.5	99.7	133.0	166.2	199.4	232.7	265.9	299.1
Storage Duration (Minutes)	10	15	20	25	30	35	40	45

Steam Buffer Storage:

This storage system has to be provided in the steam transport line before steam engine. The drawback of this type of storage is the large storage volume at high temperature and pressure. The thermal capacity of the storage system will be very high due to thicker wall of the storage container from mechanical considerations and high insulation thickness for minimizing the thermal loss. An estimation of storage volume for 10 minutes duration has been made below:

Steam temperature	=	400°C
Steam pressure	=	60 ata
Specific volume of steam	=	0.048 m^3/kg
Mass flowrate of steam	=	180 kg/hr
Volume of steam for 10 minutes storage	=	180 x 0.048 x 10/60
	=	1.44 m^3
For cylindrical container consider height to diameter ration	=	1.2
Therefore, diameter	=	$(1.44 \times \pi/(4\times1.2))^{1/3}$
	=	1.15m
Height	=	1.2 x 1.15
	=	1.38m
Surface area	=	$\pi \times 1.15 \times 1.38 + 2 \times \pi \times (1.15)^2/4$
	=	7.06 m^2

A suitable alternative to the steam storage is under consideration, to have more storage capacity with less surface area for minimizing thermal loss.

5.6 COLLECTOR FIELD LAYOUT & PIPE SIZING

A well designed pipe field with collectors is extremely important in a solar system to minimize the thermal loss, thermal capacity and pumping power. To fullfil the above objectives the pipe field should have

i) minimum length of piping
ii) reduced up and down lengths
iii) reduced pipe diameters
iv) tapered branches
v) minimum possible wall thickness, and
vi) balanced fluid flow

Reduction in the piping length demands minimum possible packing factor, besides shortest laying of the pipes. The decrease in the pipe diameter increases the pumping power. Thus to obtain an optimum pipe diameter, one should have a balance between the saving of the collector area due to reduction in thermal loss and the collector area required for the increase in pumping power requirement. The piping should have a tapered section spacially on the branching, to have a uniform flow at all the sections. The tubes are preferred than the pipes to have a reduction in wall thickness, withstanding the same pressure and temperature. The collector field should be balanced with respect to the flow so that valves will not be required for maintaining a proper flow in the branch piping.

Considering the field factor for 8 m diameter dishes at the selected site to have no shadow from 8 am daily for the whole year, one gets the center to center distance between the two dishes, both in north-south and east-west directions, as 15 m. This gives packing factor on rectangular configuration as

$$\pi 8^2/(4 \times 15 \times 15) = 0.22$$

The length of downcommer or riser is approximately 12 m for the polar mounted dishes and 10 m for the flat heliodish. Two field layouts are considered as shown in Figures 19 and 20. The location for the installation of the power conversion system has been chosen relative to the dish position so as to minimize the pipe lengths.

Both the options use same length of the feed water piping, but the Option I uses 15 m more steam piping than the Option II. The Option II uses tapered piping scheme of different diameters and many bends and tees. This necessitates various insulation thicknesses appropriate for the pipe diameters. The Option I uses a single diameter pipe, respectively for feed water and steam flow, the diameters of the piping being equal to the lowest size of the Option II arrangement. Further, this scheme does not use any tee joint. For the present system the Option I will be used as the collector field layout.

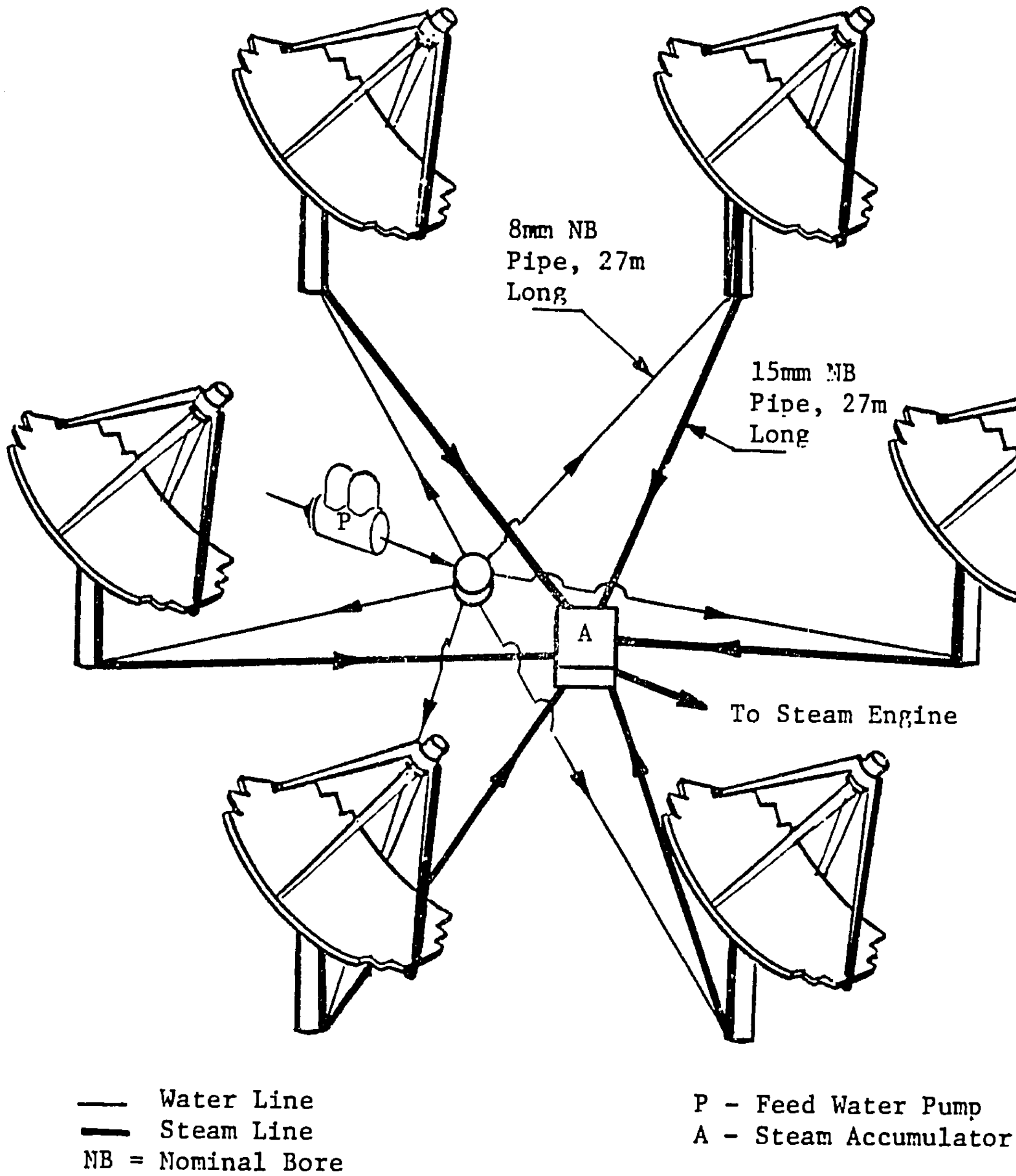


Figure 19 (A). COLLECTOR FIELD LAYOUT WITH POLAR MOUNTED PARABOLIC DISHES (OPTION I)

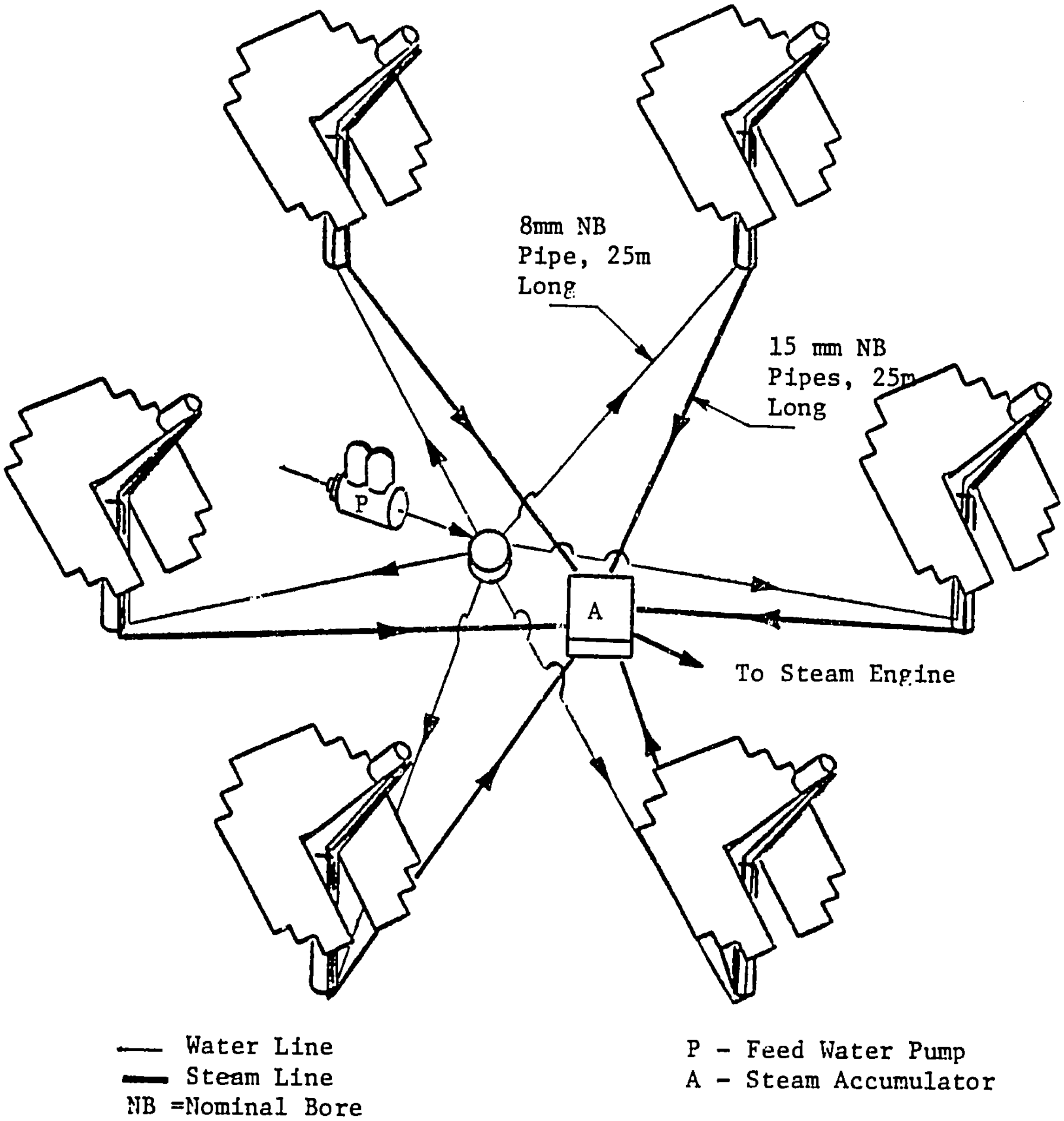


Figure 19 (B). COLLECTOR FIELD LAYOUT WITH FLAT HELIODISHES (OPTION I)

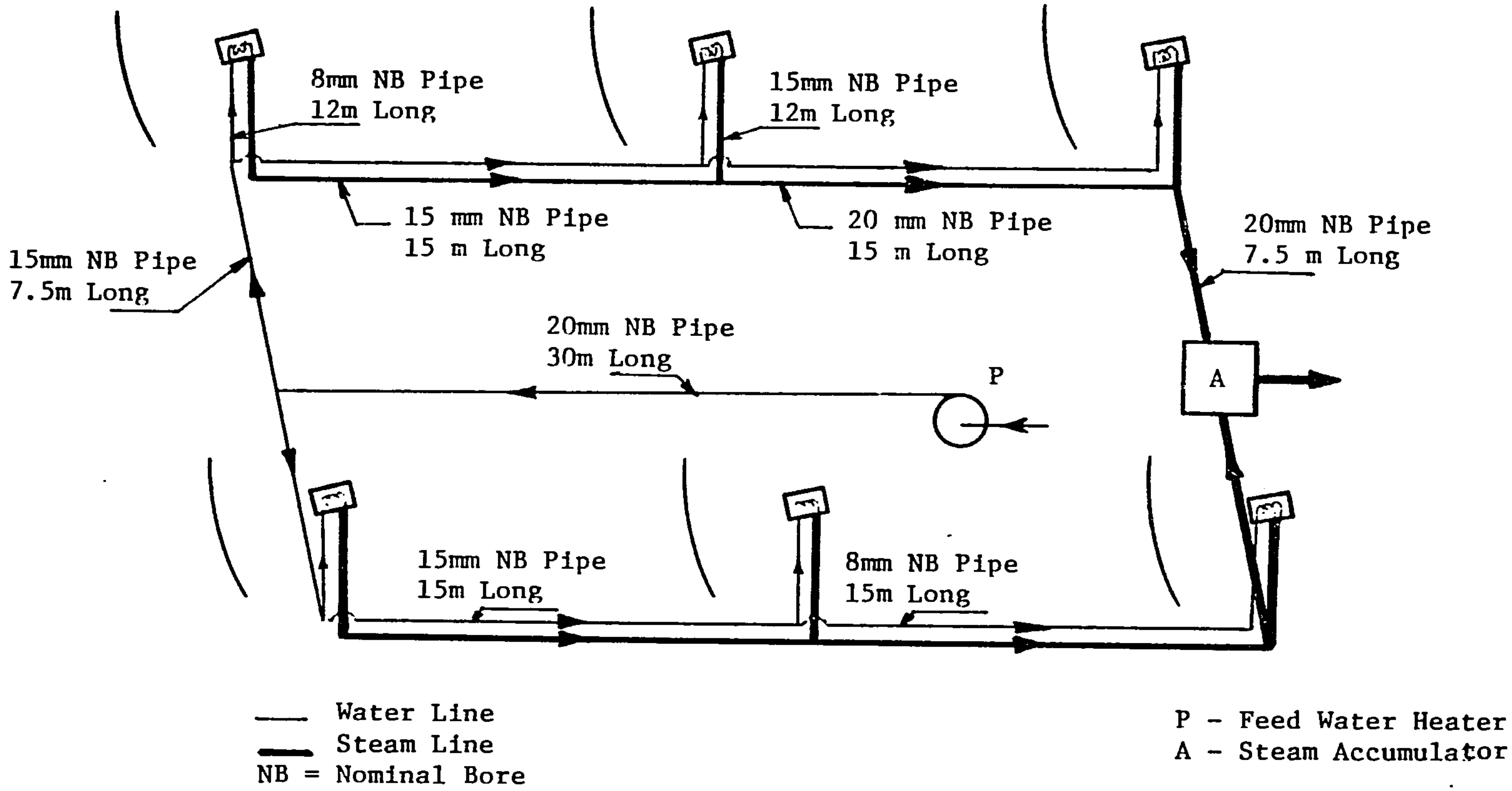


Figure 20. COLLECTOR FIELD LAYOUT (OPTION II)

A crude analysis was made to evaluate minimum diameter of pipes to have same flow at each section. The diameter of the pipes, obtained through this analysis, has been indicated in the respective figures. Option I uses 8 mm NB pipe for feed water flow and 15 mm NB pipe for steam flow. However, a more rigorous analysis on the sizing of pipes is in the process.

5.7 DETERMINATION INSULATION THICKNESS

The thermal loss from a solar power plant system should be as minimum as possible in order to decrease the collector area, which is the highest cost sharing component in the whole system. Therefore, analysis of thermal loss and determination of the optimum insulation thickness is quite important specifically for the present system incorporating steam transportation, where temperature drop will be large even with very small thermal loss. The critical sites for the thermal losses are, steam piping, valves, pipe supports, steam accumulator/buffer storage, and steam engine.

After having an optimal piping system, one has to emphasize on the following points to reduce heat loss:

i) Lower conductivity insulation
ii) Greater insulation thickness
iii) Nesting, and,
iv) Reduction in thermal shyphoning

Table 10 gives (3) the various candidate insulation materials with their properties suitable for the present operating temperatures. For high temperature insulation kaowool and for low temperature insulation fibre glass has been chosen. The higher the thickness of insulation, more will be the thermal capacity. Thus, one should keep this point in mind, while optimizing the thickness of insulation. The optimum insulation is the one where cost of addition of insulation equals the cost of saving of the collector area, because cost of the other components in the system remains unaffected. Due to high temperature difference between feed water and steam, nesting is not possible for single loop steam Rankine system. Thermal shyphoning can be reduced by laying the pipes either at an angle or vertical position as required, suiting to thermal energy storage location. However, in the present system this is not critical. While a detailed analysis is being made, the crude estimation of insulation thickness of only transport piping results in the following:

Table 10. Selected Insulation Materials

	Thermo 12	Foam Glass	Certain Teed	Kaowool	Owens Corning	Micro Quartz
1. Thermal Conductivity Kcal/m^2/hr °C/m (at 190°C)	0.06	0.057	0.052	0.05	0.062	0.053
2. Density kg/m^3	208.3	136.2	84.1	48.1	96.1	48.1
3. Specific heat Kcal/kg°C	0.20	0.18	0.2	0.244	∿0.2	∿ 0.2
4. Maximum Temperature °C	831.0	468	470	1245	359	1109
5. Performed	Pipe	Pipe	Pipe	Blanket	Pipe	Felt
6. Material	Calcium Silicate	Glass	Fibre-glass with Resin	Alumina Silicate	Fibre-glass with Resin	Silica
7. Manufacturer	Johns Mans-field	Pitts-burg Corning	Certain Teed	Babcock & Wilcox	Owens Corning	Johns Mans-field

Steam Piping:

Considering the steam pipes be at 2m height from ground level, the length of piping for polar mounted dishes = 25 m

OD of steam pipe = 19 mm
Steam flow = 30 kg/hr

Steam condition at receiver outlet:

Temperature = 500°C
Pressure = 65 ata (approx.)
Heat content of steam = 27.2 kW_t

Steam condition at engine inlet:

Temperature = 400°C
Pressure = 60 ata

Ambient temperature = 30°C
Average temperature of steam = 450°C
Thermal conductivity of insulation at 450°C = 0.07 kcal/m^2°C/m
Insulation thickness = 150 mm (considered)

The analytical relationship for heat transfer considering tube size, insulation thickness, conductivity, temperature and outside film coefficient is given by the equation,

$$q = 2\pi KL \, \Delta T/(\ln(r_o/r) + (K/r_o h_o))$$

where $k/r_o h_o$	=	outer surface conductance due to convection
K	=	conductivity of insulation
L	=	length of piping
ΔT	=	temperature difference between fluid and ambient
r	=	outer radius of pipe
r_o	=	outer radius of insulation
h_o	=	outside film coefficient
q	=	heat transfer rate

At an outside film temperature of 95°C, and air velocity 10 Kmph, using equation 10.2 of reference (7), one gets,

$$h_o = 12.69 \text{ kcal/m}^2\text{hr°C}$$

Therefore, due to low value of K and the product $h_0 r_0$ being large, the term $(K/h_0 r_0)$ becomes negligible in comparison with the term $\ln(r_0/r)$.

Thus heat flow, $q = 2\pi KL\Delta T/\ln(r_0/r)$
$= 2\pi \times 0.07 \times 25(450-30)/\ln(159.5/9.5)$
$= 1637.2$ kcal/hr
$= 1.9$ kw_t

Thermal loss = 1.9/27.2 = 7%

Enthalpy drop in steam

= 1637.2/30 = 54.57 kcal/kg.

The enthalpy drop corresponds to a temperature drop of 95°C (approx.)

Steam Accumulator:

Capacity of storage for 10 minutes

= 180 x 10/60 = 30 kg.

Temperature of steam = 420°C (considered)
Pressure of steam = 60 ata
Insulation thickness = 0.4m (considered)
Diameter of tank = 1.115m
Height of tank = 1.38m
Mean radius of insolation = 0.775m
Radius of tank = 0.575m
Mean height of insulation = 1.78m

Heat loss from cylindrical surface
$= 2\pi \times 0.07 \times 1.78\ (420-30)/\ln(0.775/0.575)$
$= 1022.9$ kcal/hr
$= 1.2$ kW_t

Heat loss from bottom and top surface
$= 2\pi(0.775)^2 \times 0.07\ (420-30)/0.4$
$= 257.6$ Kcal/hr
$= 0.3$ kW_t

Total heat loss = 1.5 kW_t

Feed Water Piping:

Outside diameter of pipe = 13mm
Length of pipe = 25m
Insulation thickness = 75 mm (considered)
Temperature of water = 150°C
Thermal conductivity of
fibre glass insulation = 0.05 $Kcal/m^2 \ hr°C/m$

Heat flow = $2\pi \times 0.05 \times 25 \ (150-30)/\ln(81.5/6.5)$
= 372.7 Kcal/hr
= 0.4 kW_t

Enthalpy Drop = 372.7/30 = 12.4 Kcal/kg

Temperature drop = 12°C (approx.)

5.8 Pump Sizing and Auxiliary Power Requirement

Power Sizing:

After completion of the design of the receiver coil and complete piping system with different valves and fittings, it will be possible to estimate the pressure drop in the system required for sizing the feed water pump. Similarly condenser pump capacity can be determined after knowning the specifications of the cooling tower, and the piping details along with fittings.

Auxiliary Power:

Once the capacities of various pumps are determined, and specifications of control components and tracking electronics are drawn, the total auxiliary power requirement can be known. On the basis of this power requirement, the capacity of the biogas engine/generator assembly and the plant can be determined.

5.9 Thermal Capacity and Hours of Operation of the System

To determine the thermal capacity, one has to know the mass and specific heat of each material contained in all the components/subsystems responsible for heat transmission. This includes the insolation added for minimizing the thermal losses. Using heat capacity values a transient equation can be developed in terms of temperature, insulation, and ambient temperature. This will determine the time required to heat up the system and to supply the steam at the engine inlet, at the required temperature and pressure for delivering power. On the basis of this analysis, hours of operation of the plant can be determined.

5.10 Control System (Preliminary)

The control system for the present solar power plant should take care of the following important aspects:

i) Initial heating up period

ii) Concentrator tracking control

iii) Controls of various critical parameters

iv) Power output at constant voltage and frequency

v) Constant power output or variable power output with proper matching of loads

vi) Safety control, and

vii) Start up and shutdown sequences

A suitable control system will be developed after finalization of the functional requirements of the total system.

6.0 CONCLUSION

The overall system design for a solar thermal power plant has been made, keeping in view the available technologies which are suitable for rural applications. The power plant uses, six point focusing collectors of 8 m diameter each with faceted mirrors reflective surface. The system is based on a single loop, steam Rankine thermodynamic cycle using a steam engine/generator assembly on a central power conversion approach. The power plant is expected to deliver 12.4 and 19.5 kW_e, respectively at 0.55 and 0.8 kW/m^2 insolation level.

Sufficient inputs have been provided for detailed design of the subsystems/components in the total system. The thermal loss through steam transportation seems to be critical which requires special attention. This being a first system of its kind, many assumptions have been made subject to verification while testing the subsystems and the complete system.

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APPENDIX A

THERMAL DESIGN OF CONDENSER

Hot Fluid: Steam

Inlet Condition: Pressure 6 ata
Temperture 157°C
Enthalpy 645 kcal/kg
Enthalpy of saturated liquid 161 kcal/kg

Outlet Condition: Pressure 6 ata (approx.)
Temperature 150°C
Subcooling 7°C
Enthalpy 152 kcal/kg
Mass flow 180 kg/hr.

Cold Fluid: Water

Inlet Temperature: 33°C (91.4°F)
Outlet Temperature: 37°C (98.6°F)

Heat Load

Condensation = (645-161) 180 = 87120 kcal/hr.
= 345714.3 Btu/hr.
Subcooling = (161-152) 180 = 1620 kcal/hr.
= 6428.6 Btu/hr.

Cooling Water Requirement

Total heat to be transfered
= 87120 + 1620 = 88740 kcal/hr.
Temperature rise = 37-33 = 4°C
Mass flow rate of cooling water = 88740/4 = 22185 kg/hr.
= 48895.7 lb/hr.
= 48895.7/(3600 x 62.4)
= 0.22 ft^3/sec

Minimum water velocity set is 5 ft/sec.
Therefore, tube cross-sectional flow area = 0.22/5 = 0.044 ft^2.

Characteristics of Tubing

OD of Tubing Inch	SWG Gauge	Thickness Inch	Internal Flow Area in^2	External Surface Area ft^2/ft	ID of Tubing inch	$\frac{OD}{ID}$
1/2	16	0.065	0.1075	0.1309	0.37	1.351
3/4	14	0.083	0.2679	0.1963	0.584	1.284

Assume 6" ID shell, single pass, with 24 tubes of 3/4" OD, 2.25 ft long on 1" triangular pitch.

Now velocity in each tube

= 0.22/(24 x 0.2679/144)

= 4.9 ft/sec.

Total surface area available allowing 3" thickness for tube sheets.

= 24 x 0.1963 x 2.0 = 9.4 ft^2

Logmean Temperature Difference

Water flow rate = 48895.7 lb/hr.

Temperature of water at the end of condensation

= 98.6- (345714.3/48895.7)

= 91.53°F

Condensing: 157°C ------- 157°C
(314.6°F) (314.6°F)
37°C ------- 91.53°F
(98.6°F)

LMTD = ((314.6 - 98.6) - (314.6 - 91.53))/
ln((314.6 - 98.6)/(314.6 - 91.53))
= 219.52°F

No correction required for condensing section.

Subcooling: 157°C -------- 150°C
(314.6°F) (302°F)
91.53°F ------- 33°C
(91.4°F)

LMTD = ((314.6 - 91.53) - (302 - 91.4))/
ln ((314.6 - 91.53)/(302 - 91.4))
= 216.78°F

No correction required.

Tube Side Film Coefficient

From Figure 10.40[6] for velocity of 4.9 ft/sec and average temperature of 95°F, heat transfer coefficient = 1280 Btu/hr ft^2°F. Correction factor for 3/4" tube = 1.02.

Therefore, tube side film coefficient

= 1280 x 1.02 = 1305.6 Btu/hr ft^2°F.

Shell Side Film Coefficient

The subcooling load is 1.8% of the condensing load. Hence complete 2 ft length has been considered for condensation.

Tube Loading, $G_o = M/(LN^{2/3})$
$= (180 \times 2.204)/(2 \times (24)^{2/3})$
$= 23.84$ lb/hr ft

Properties of film(7) at temperature,
$t_f = 314.6 - 3/4\ (314.6 - (98.6 + 91.53)/2) = 150°F$,
$\mu_f = 1.055$ lb/hr ft
$\rho_f = 61.2$ lb/ft^3
$k_f = 0.382$ Btu/hr ft°F

Condensation film coefficient
$= 0.945\ (k_f^3 \rho_f^2 g/\mu_f G_o)^{1/3}$
$= 0.945\ (((0.382)^3 (61.2)^2 32.2(3600)^2)/(1.055 \times 23.84))^{1/3}$
$= 1429.87$ Btu/ft^{2}hr°F

Overall Heat Transfer Coefficient

Tube side fouling factor = 0.001 ft^{2}hr°F/Btu
Shell side fouling factor = 0.001 ft^{2}hr°F/Btu

Overall heat transfer coefficient,

$$\frac{1}{U} = 1/1429.87 + 0.001 + 0.001 \times 1.284 + 1.284/1305.6$$

$$= 3.9668 \times 10^{-3} \text{ ft}^2\text{hr°F/BTU}$$

Therefore, U = 252.09 Btu/ft^{2}hr°F

Condensing area = 345714.3/(219.52 x 252.09)
= 6.25 ft^2

Subcooling Area

There is no necessity for subcooling area calculation as 9.4 ft^2 is sufficient to take care of that.

Therefore, total heat transfer area for condenser = 9.4 ft^2

Baffles

Baffle pitch = 3"
Baffle cut = 0.25%

Nozzle Sizes

Water connections:

Flow rate = 48895.7 lb/hr = 0.22 ft^3/sec.
Maximum allowable velocity = 7 ft/sec.
Design for 0.22 x 1.5 = 0.33 ft^3/sec.
Area required = 0.33/6 = 0.055 ft^2
Diameter = $(0.055 \times 144 \times 4/\pi)^{0.25}$
= 3.17" (3" adopted)
Velocity = $0.22 \times 4 \times 144/\pi \times 9$
= 4.5'/sec.
Nozzle size = 3" ID pipe

Condensed water outlet:
Mass flow rate = 180 x 2.204 = 396.7 lb/hr.
= 396.7/(3600 x 57.2)
= 1.9264×10^{-3} ft^3/sec.
Design for $1.9264 \times 10^{-3} \times 1.5 = 2.8897 \times 10^{-3}$ ft^3/sec.
Considering 3/8" ID pipe, the velocity
= $2.8897 \times 144 \times 4 \times 64/\pi \times 9 \times 1000$
= 3.8 ft/sec.
This value is well within maximum allowable velocity.

Nozzle side = 3/8" ID pipe

Steam inlet:
Mass flow rate = 396.7 lb/hr.
Design rate = 396.7 x 1.5 = 595 lb/hr.
Pressure = 6 ata = 88.5 psia

From figure 10.53, at 88.5 psia and 18 molecular weight, the maximum suggested vapour velocity through a nozzle is

49 x 1.2 = 58.8 ft/sec.

Specific volume = 5.2 ft^3/lb.
Steam flow = 396.7 x 5.2/3600 = 0.5738 ft^3/sec.

Considering 1/2" ID pipe, the velocity
= $0.5738 \times 144 \times 4 \times 4/\pi$ = 420.8 ft/sec.

Considering 2" ID pipe, the velocity
= $0.5738 \times 144 \times 4/\pi \times 4$ = 26.3 ft/sec.

Nozzle size = 2" ID pipe

Pressure Drop:

Tube side:
Water flow rate = 48895.7 lb/hr
= 48895.7/24 = 2037.3 lb/hr/tube

From Figure 10.99, at water flow rate of 2037.3 lb/hr/tube for 3/4" OD 14 BWG tube, pressure drop = (8.5/100) psi/ft.

Therefore, total tube pressure drop
= (8.5/100) x 1 x 2.25
= 0.19 psi

From Figure 10.100, at tube velocity of 4.9 ft/sec and specific gravity 1.0, pressure drop = 0.65 psi/pass.

For single pass case entry drop
= 0.65 psi

The effect of water temperature has not been considered.
Therefore, total pressure drop = 0.19 + 0.65
= 0.84 psi
Pressure drop considered = 1.5 psi.

Shell Side:

The pressure drop in the shell side is negligible and hence not calculated.